Engineering Science Series

ENGINES AND BOILERS

ENGINEERING SCIENCE SERIES

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ENGINES AND BOILERS

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PREFACE

This text book on Engines and Boilers is intended for use in engineering schools which offer an elementary course in Heat Engines. No attempt has been made to cover the more advanced work in Thermodynamics, or to give an exhaustive treatment of the subject of Heat Power.

This work is the result of the author's experience during the several years that he taught classes in Engines and Boilers and in allied subjects at Purdue University. Much of the material was given to the students first in lectures, and later in the form of mimeographed notes. It is now presented in book form with the hope that it may be of value in other engineering schools.

At the end of the book a list of representative problems is given. It has been the author's experience that the student obtains a better understanding of the subject if he is required to work problems related to the matter in the text.

The author wishes to thank Professor C. H. Lawrence for valuable suggestions made in regard to the form of presentation of some of the work.

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ENGINES AND BOILERS

CHAPTER I

PRESSURE, TEMPERATURE AND HEAT UNITS

1. Pressure Units. — In steam engineering, pressure is measured in the following units:

(1) In pounds per square inch.
(2) In inches of mercury.
(3) In inches of water.

In this country boiler pressures are ordinarily measured in pounds per square inch above atmospheric pressure. Condenser pressures are commonly measured from atmospheric pressure in inches of mercury, *i.e.* the difference between the pressure in the condenser and atmospheric pressure is read on a mercury column. Draft pressures are usually measured in inches of water. Pressure gages and vacuum gages are so constructed that they read zero at atmospheric pressure. The atmosphere exerts a variable pressure, which is about 14.5 pounds per square inch at ordinary altitudes and under ordinary conditions. At sea-level, the standard is taken as 14.7 pounds per square inch, which is equivalent to 29.92 inches of mercury.

As the atmospheric pressure is slightly variable, it is necessary in accurate work that pressures be reduced to an *absolute* basis. Since the zero reading of a boiler pressure gage means atmospheric pressure, the *absolute* pressure will be the sum of the gage pressure and atmospheric pressure.

A partial vacuum usually exists in a condenser, since the absolute condenser pressure is usually less than atmospheric pressure. Since the vacuum gage as well as the boiler pressure gage reads zero at atmospheric pressure, the *absolute* pressure in the condenser is the *difference* between the atmospheric pressure and the vacuum-gage pressure.

Figure 1 shows diagrammatically the relation between these pressures. In this figure, a is the boiler-gage pressure, b the

atmospheric pressure, and c the *absolute* boiler pressure. Likewise, d represents the vacuum-gage pressure, which is measured downward from the atmospheric pressure; and e, the difference between b and d, is the absolute condenser pressure.

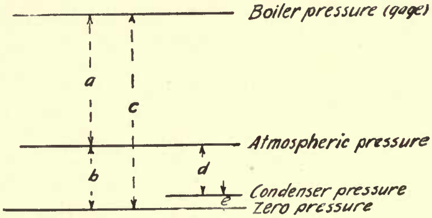


FIG. 1

Barometers used in engineering work in this country are usually graduated in inches. The mercury barometer is used because mercury is the most convenient fluid for this use. A cubic inch of mercury weighs very nearly 0.49 of a pound. Hence the pressure in pounds per square inch is the barometric reading in inches multiplied by 0.49 Vacuum gages also are commonly graduated to read in inches of mercury. Therefore the absolute condenser pressure in pounds per square inch is 0.49 times the difference between the barometer and the vacuum-gage readings.

EXAMPLE 1. Find the absolute boiler pressure when the pressure gage reads 110 pounds and the barometer reads 29.4 inches.

SOLUTION. When the barometer reads 29.4 inches, the atmospheric pressure is $29.4 \times 0.49 = 14.4$ pounds per square inch. The absolute boiler pressure is then $14.4 + 110 = 124.4$ pounds per square inch.

EXAMPLE 2. Find the absolute pressure in a condenser when the barometer reads 29.8 inches and the vacuum gage reads 27.3 inches.

SOLUTION. The absolute condenser pressure is the difference between the barometric and the vacuum-gage pressures, which in inches of mercury is $29.8 - 27.3 = 2.5$. This reduced to pounds per square inch is $2.5. \times 49 = 1.22$.

2. Temperature Units. — Ordinary temperatures are measured by means of the mercury thermometer. For higher temperatures, such as those that occur in furnaces, special thermometric devices called ***pyrometers*** are used.

Three thermometer scales are in use, the ***Fahrenheit***, the ***Centigrade***, and the ***Réaumur***. In the Fahrenheit scale, the difference between the temperatures of melting ice and boiling water at sea level is divided into 180 divisions or degrees; the freezing point is 32° and the boiling point 212°. This makes the zero

point come 32° below the freezing point of water. In the Centigrade scale, the freezing point is 0° and the boiling point is 100°. In the Réaumur scale, the freezing point is 0° and the boiling point is 80°. Figure 2 shows graphically the relation between these scales. It is readily seen how the reading on one scale may be reduced to that of either of the others. It is serviceable to remember that a difference of temperature equal to 5° on the Centigrade scale is equal to 9° on the Fahrenheit scale.

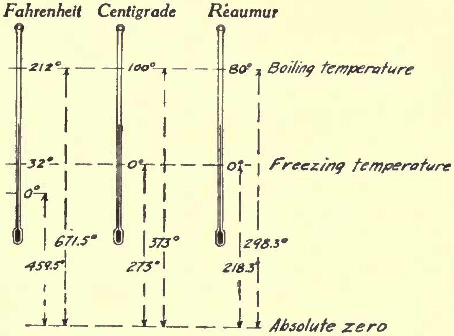


Fig. 2

Experiment shows that a perfect gas under constant pressure at 32° Fahrenheit expands 1/491.5 part of its own volume for each degree (F.) that its temperature is increased. Because of this we call the point 491.5° below the freezing point, or −459.5° on the Fahrenheit scale, the ***absolute zero.*** This corresponds to a Centigrade temperature of −273°.

3. Heat Units. — In engineering work in this country, it is customary to use English units. Our common heat unit is the ***British thermal unit*** (***B.t.u.***), which is practically defined as the amount of heat necessary to raise the temperature of one pound of pure water from 62° to 63° Fahrenheit. In the metric system the engineers' heat unit is the large ***calorie,*** which is the amount of heat necessary to raise the temperature of a kilogram of water from 15° to 16° Centigrade. As one kilogram = 2.2046 pounds, and as 1° Centigrade = 1.8 degrees Fahrenheit,

one calorie = 3.968 B.t.u., or one B.t.u. = 0.252 calorie.

4. Mechanical Equivalent of Heat. — Heat experiments have been made to determine the relation between heat-energy and mechanical work. The latest and most refined experiments show that *one B.t.u. as above defined is substantially equivalent to* 778 *foot-pounds of work.* This relation is called the ***mechanical equivalent of heat.***[1]

[1] For definition of the "mean B. t. u." and corresponding Mechanical Equivalent of Heat see A. S. M. E. Power Test Code, Edition of 1915, p. 28.

CHAPTER II

FUEL

5. Introduction. — The source of the world's supply of energy is the sun. In the use of water power we are drawing, in point of time, almost directly from the sun. On the other hand, in the use of such fuels as coal, gas, and oil, we make use of a store of energy that has been accumulating for ages.

While the world's supply of coal, oil, and gas is limited, we have used but a very small part of the known deposits. The past century has seen a marvelous change in our manner of living and in our ways of thinking. Our vast commercial system with its perplexing problems has arisen during the past few generations. One of the chief causes of this great change is our ability to utilize the vast stores of energy to be found in nature. The steam-engine has been the chief means by which the energy stored in our coal deposits has been tapped and forced to do the work of man. What will become of this modern civilization of ours when the fuel supply upon which it is based is exhausted, is an interesting problem of the future. Already conservationists are calling to us to stop the great waste of our natural resources.

Of all our fuels, ***coal*** is the most important. Coal is the remains of vegetable matter deposited in remote geological ages. It is well known that wood rots but little when kept under water. If the water be fresh, the wood is not eaten by the teredo worm or other forms of aquatic life, and will be kept in a fair state of preservation for thousands of years. If tree trunks and other vegetable matter fall into a fresh-water swamp and are submerged before they rot, and if this continues for many centuries, there will be a great accumulation of it. Our coal deposits are the result of such an accumulation of vegetable matter. Under tropical conditions accompanied by a large supply of carbon dioxide in the atmosphere the growth was very rapid and a deep bed would collect in a comparatively short time.

Geologists tell us that in the past, parts of the surface of the earth have gradually risen while others have fallen. At a remote time, the tops of our highest mountains may have been the bottom of the sea. Suppose a former swamp with its accumulated vegetable matter is now sunk, and that great quantities of silt or

other material are deposited upon it. The weight of the material above will compress the vegetable matter into a compact and dense mass. It is also possible that it will be subject to a high temperature, which will change its chemical composition. Varying conditions of pressure and heat are thought to be responsible largely for the differences between the various kinds of coal.

The principal constituents of coal are carbon, hydrogen, oxygen, nitrogen, sulphur, and refractory earths called ash. The wood-fiber of the original vegetable matter was composed chiefly of hydrocarbons. While under the influence of great pressure, it has at some period of its history been subjected to considerable heat and therefore undergone a process of destructive distillization. This has driven off much of the volatile matter of the original vegetable material and left a considerable portion of uncombined carbon, which is called ***fixed carbon.*** The remainder of the carbon exists in combination with hydrogen. These carbon and hydrogen compounds are called ***hydrocarbons.*** They are easily volatilized, and so comprise a part of the volatile matter of the coal.

Oxygen and hydrogen are always present in coal in the form of water. This water is volatilized, of course, when the coal is burned. Since heat is required to evaporate and to superheat it, water is a detriment if present in too large a quantity. A small amount of water, however, seems to aid in the combustion of some coals.

All coal then contains fixed carbon, volatile matter (hydrocarbons and water), and ash; and it may contain other substances (*e.g.*, sulphur). The combustible is the fixed carbon, the hydrocarbons, and part of the sulphur that may be present. Excessive sulphur is undesirable because it is harmful to the metal of the boiler and the stack if moisture is present, since it may form sulphurous or sulphuric acid; it combines with the ash to form a fusible slag or clinker, which is commonly objectionable; and it makes the fuel more liable to spontaneous combustion when stored in deep piles.

Coals that have been subjected to the greatest pressure and heat are composed mostly of fixed carbon, and contain only a small amount of volatile hydrocarbons. Such coals are called ***anthracite.*** Coals containing larger quantities of volatile hydrocarbons are called ***bituminous.*** Since there is no definite divid-

ing line between these two classes, but the two seemingly overlap, the terms ***semi-anthracite*** and ***semi-bituminous*** are commonly used to designate coals to which are ascribed certain properties of each class.

6. Anthracite Coal. — Anthracite coal contains but a small amount of combustible volatile matter. While it is considered better for some uses, its heating value is less than that of good grades of bituminous coal. It burns slowly, with but a small flame and practically no smoke. Due to its slow burning qualities, a relatively large grate area is needed on which to burn it. The supply of this coal that is easily obtained has been diminishing in this country, and its demand for domestic use has greatly increased during the past few years. This has led to a rapid increase in price and a great diminution of its use for power purposes.

Anthracite is considered much superior to bituminous coal for the production of producer gas. This is due to the fact that it is so free from the hydrocarbons that produce tars. The formation of tar has been the great objection to the use of bituminous coals in the producer plant.

Average anthracite coal contains about 85% fixed carbon, 4% volatile matter, 9% ash, and 2% water. The heat value averages about 13000 B.t.u. per pound.

7. Bituminous Coal. — Most of the coal used for power purposes is bituminous. This coal contains a larger percentage of volatile combustible matter. It burns at a lower temperature than does anthracite, and with a much longer flame. The length of the flame varies with the composition, some kinds being called ***long-flaming*** and others ***short-flaming.*** The ordinary furnace is not usually so constructed as to give the volatile hydrocarbons a chance to be completely burned. This results in the formation of smoke. Engineers have spent much time and study on the prevention of smoke. Upon yielding up their volatile matter some coals fuse and form a solid mass or cake of the nature of coke. These are called ***caking coals.*** This action hinders the draft. If rapid combustion is desired, the mass must be broken up.

The composition of bituminous coal varies greatly, but the average of the better grades may be taken as 65% fixed carbon, 28% volatile combustible, 5% ash, and 2% moisture, with a heat value of 14000 B.t.u. per pound.

8. Lignite and Peat. — ***Lignite*** or ***brown coal*** is high in volatile combustible and also contains much moisture. The evaporation of this moisture after mining causes the lignite to crumble or slack. It is usually inferior to anthracite and bituminous coals, but it is used where it is easily obtained and where better coal is expensive. Abroad, lignite is often formed into briquettes.

Peat is the partly decayed remains of vegetation that accumulates in bogs. While it is an inferior fuel, it is used to a considerable extent abroad. It is sometimes pressed into briquettes.

9. Natural Gas. — ***Natural gas*** is used to some extent for power purposes in sections of the country within reach of the gas fields. It contains about 90% marsh gas (CH_4) and has a heat value of nearly 1000 B.t.u. per cubic foot.

10. Oil. — ***Crude petroleum*** and ***fuel oil*** are used to a considerable extent in parts of this country. The petroleum produced in the eastern and middle states is of a paraffin base, while that from Texas and California is of an asphalt base. Gasoline and other light oils are distilled from petroleum, and the residue is sold as *fuel oil.* Petroleum is composed principally of hydrogen and carbon in the form of hydrocarbons and has a heat value of about 20000 B.t.u. per pound.

11. Coal Fields of the United States. — Our principal deposits of anthracite coal are in eastern Pennsylvania. The deposits are not of large extent and the best are rapidly becoming exhausted. It is claimed that there is anthracite in Alaska. Only a small proportion of the anthracite mined is being used for power purposes, the rest going for domestic heating and like purposes.

The best of our bituminous and semi-bituminous coal is taken from the field that includes western Pennsylvania, eastern Ohio, a large part of West Virginia, and eastern Kentucky. Southwestern Indiana and most of the state of Illinois are underlaid with coal of a fair quality. There is also a field running north from Oklahoma through eastern Kansas and western Missouri into Iowa. The coal from the latter is generally of poor quality, and is used only locally. Immense coal fields exist in southern Utah and Colorado, and in New Mexico and Arizona. No complete survey of these fields has been made as yet, and they have not been developed up to the present. There is also a coal and lignite field in eastern Montana and western North Dakota.

The peat beds of the country are principally in Minnesota, Wisconsin, Michigan, New York, and the New England states.

Oil is produced in the territory occupied by the eastern coal fields, in Kansas, Oklahoma, Texas, California, and, to a smaller extent, elsewhere.

12. Coal Storage. — In the operation of most steam-power plants, it is essential that a constant supply of coal be available. Owing to unsettled labor conditions at the mines and to uncertain transportation facilities, it is necessary that there be some storage capacity. With anthracite coal this is a simple problem, but it is not so with certain grades of bituminous coal. Upon exposure of bituminous coal to the air, there is a considerable oxidation of the hydrocarbons with attendant heat production. If the coal pile is large, this heat may start a fire which is costly and hard to extinguish. The origin of such a fire is called *spontaneous combustion*. Even if fire does not start, there is a loss of heat-value up to as high as 10% in some grades of coal. If the moisture content is large, the weathering is accompanied by a crumbling or slacking. Storage piles are often ventilated in order to keep them cool. In some large plants, the storage is so arranged that it may be submerged in water. This obviates the fire risk, and reduces the other losses to a minimum.

13. Determination of Heating Values of Fuel. — In plants where large amounts of fuel are used, it is quite common to buy coal on the basis of its heating value. In accurate tests of power plants it is also necessary to know the heating value of the coal used. Care must be exercised in order to get a fair sample of the fuel. The heating value may be determined in two ways, by combustion in a calorimeter, or by chemical analysis.

In the ***calorimeter method*** a sample of the fuel is placed in a steel bomb along with compressed oxygen. The bomb is placed in a calorimeter containing water, and the fuel is ignited by means of an electrically heated wire. Upon the firing of the fuel, heat is given to the water. It is possible to determine from the rise of the temperature of the water the amount of heat generated. The value thus determined is known as the ***higher calorific value.*** It must be remembered that it is seldom possible actually to get this amount of heat by burning in a furnace. This is due to the fact that the hydrogen in the fuel combines with the oxygen of

the air to form water, which ordinarily passes from the furnace in the form of steam, carrying with it the heat of vaporization of the steam. The *lower heat-value* does not contain this heat of vaporization. The *higher value* is the accepted standard.

There are two methods of ***chemical analysis,*** the ***ultimate,*** and the ***proximate.*** The ultimate analysis may be made on the basis either of moist or dry fuel. The latter is commonly accepted. If the analysis is made on the basis of moist fuel, it may be converted to the dry basis by dividing the percentage of the various constituents by one minus the percentage of the moisture. The ultimate analysis gives the percentage by weight of carbon, hydrogen, oxygen, nitrogen, sulphur, and ash. Knowing the composition of the fuel, the heat-value may be determined by a formula. An accepted formula is a modification of that of DULONG:

$$\text{B.t.u. per pound of dry fuel} = 14600\ C + 62000\ (H - O/8) + 4000S,$$

in which C, H, O, and S represent the proportionate parts by weight of carbon, hydrogen, oxygen, and sulphur. The heat-value of pure sulphur is 4000 B.t.u. per pound, but the sulphur in coal is mostly in a form that is noncombustible. The results of the ultimate analysis agree closely with those obtained from the calorimeter. The proximate analysis gives the proportion of fixed carbon, volatile combustible, moisture, and ash. Since the heating value of the volatile combustible is not determined, the results are not as reliable as those of the previous method. It is very often used, however, because it is easily made and affords a rough comparison of various fuels.

In making the proximate analysis, a weighed sample of coal is placed in a crucible of known weight and is kept at a temperature a few degrees above the boiling point of water for an hour. From the loss of weight, the moisture in the coal is calculated. The sample is next heated in a hot flame a few minutes with the lid on the crucible. This drives off the volatile matter, and the difference in the new and previous weight is the amount driven off. Now the sample is heated for two hours with the lid off, all fixed carbon is burned out, and the weight is determined by difference as before. The weight left is that of the ash. After having found the percent by weight of the moisture, volatile matter, fixed carbon, and ash, the approximate calorific value may be found by means of the chart, shown in Fig. 3, which

has been constructed from the values determined in a large number of accurate analyses. On this chart the heat-value in B.t.u. per pound of combustible is plotted against the percent of fixed carbon in the total combustible. It is assumed that the volatile matter and the fixed carbon constitute the combustible, the ash and the moisture being noncombustible. The method of getting the heating value may be shown by examples.

EXAMPLE 1. Determine the calorific value in B.t.u. per pound of dry coal having the following ultimate analysis: carbon = 75.46%, hydrogen = 3.34%, oxygen = 2.70%, nitrogen = .53%, sulphur = 2.54%, and ash = 15.43%.

SOLUTION. The B.t.u. per pound = $14600 \times .7546 + 62000\ (.0334 - .0270/8) + 4000 \times .0254 = 12880$ B.t.u. The oxygen calorimeter gave a value of 13000 B.t.u. per pound for this same sample.

Chart for determining Heat Value of Combustible with Different Percentages of Fixed Carbon from Proximate Analysis

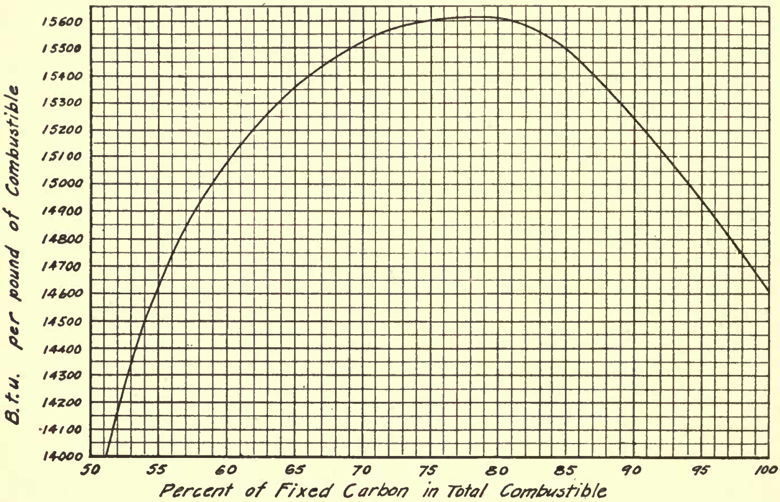


FIG. 3

EXAMPLE 2. Determine the calorific value of coal which has the following proximate analysis: moisture = 4.7%, volatile matter = 24.6%, fixed carbon = 62.5%, and ash = 8.2%.

SOLUTION. The combustible being composed of the volatile matter and fixed carbon comprises $24.6 + 62.5 = 87.1\%$ of the weight of the coal. Of this combustible the fixed carbon is $62.5/87.1 = 71.7\%$. From the chart it is seen that for 71.7% the B.t.u. per pound of combustible is 15550 B.t.u. As the combustible comprises only 87.1% of the total weight the heating value will be $.871 \times 15550 = 13550$ B.t.u. per pound of wet coal, or if reduced to the dry basis, $13550/(1.00 - .047) = 14200$ B.t.u. per pound.

14. Combustion. — By combustion is meant the rapid chemical combination of oxygen with the carbon, hydrogen, and sulphur in fuel. Combustion is complete when the maximum amount of oxygen is used in the combination. One atom of carbon will combine with one atom of oxygen to form carbon monoxide (CO). This is not complete combustion, however, for one atom of carbon will combine with two atoms of oxygen, forming carbon dioxide (CO_2), if sufficient oxygen is present.

Since the atomic weight of oxygen is 16, and that of carbon is 12, it takes 32 pounds of oxygen to 12 pounds of carbon to form carbon dioxide, *i.e.*, one pound of carbon requires for its complete oxidation 2.667 pounds of oxygen. Air, by volume, is composed of about 21% oxygen and 79% nitrogen, and by weight, of 23.15% oxygen and 76.85% nitrogen. So one pound of oxygen is contained in 4.32 pounds of air. It therefore takes $2.667 \times 4.32 = 11.55$ pounds of air for every pound of carbon burned. At ordinary room temperatures one pound of air occupies about 13.4 cubic feet, so that it requires theoretically $13.4 \times 11.55 = 155$ cubic feet of air for the complete combustion of one pound of carbon.

The hydrogen in the hydrocarbons of the coal is also combustible. A part of the sulphur present may be combustible, but it is usually present in such small amounts that it will be omitted in our present computation. As explained previously, not all of the hydrogen content of the coal is combustible, since part of it is already combined with oxygen in the form of water. Therefore the available hydrogen may be expressed as $(H - O/8)$. Since hydrogen combines with oxygen to form water, in the ratio by weight of 1 to 8, it will require 8 pounds of oxygen to burn each pound of hydrogen. Since one pound of oxygen is contained in 4.32 pounds of air, it will take $8 \times 4.32 = 34.6$ pounds of air to burn a pound of hydrogen. Hence the total weight of air required to burn a pound of coal to CO_2 and H_2O is theoretically

$$11.55\ C + 34.6\ (H - O/8),$$

where C, H and O have the same meaning as in § 13.

Since the nitrogen of the air is inert, it is of no value to the combustion. Since it passes up the stack at a higher temperature than that at which it entered the furnace, it carries away heat. Any less air than the theoretically correct amount would result in the formation of a mixture of carbon monoxide and

carbon dioxide. The heat liberated by the formation of the carbon monoxide is only 4450 B.t.u. per pound of carbon, while it is nearly 14600 B.t.u. for the formation of carbon dioxide. Hence the production of carbon monoxide in a furnace means a large loss of heat. The presence of carbon monoxide in flue gas nearly always indicates a large amount of unburned hydrocarbons and hence an even greater loss of heat. If it were possible to so distribute the air that it all came in close contact with the fuel, and also to give it time enough to combine thoroughly with the fuel, the theoretical amount of air would be sufficient. Under actual furnace conditions, however, it is found that 50% or more excess of air is needed to give complete combustion of coal. A somewhat smaller excess is needed when oil is used as a fuel, because there is better distribution of the air. The greater the amount of air passing through the furnace, the greater the amount of heat it will carry along to the stack. Hence an unnecessary excess of air is not desirable, and leads to lessened efficiency. The necessary excess depends upon the conditions of draft and fire as well as upon the kind of fuel and the type of furnace. It can only be determined by actual test.

15. Composition of Flue Gas. — As just explained, an excess of air is needed in order to get complete combustion of the fuel. If it were possible to get complete combustion without this excess, our flue gas would be composed chiefly of nitrogen, carbon dioxide, and water vapor. Due to the excess of air, there will be free oxygen present in the flue gas. If there is an insufficient excess of air there will also be carbon monoxide and probably some hydrocarbons present. We have seen that the presence of carbon monoxide indicates incomplete combustion and therefore low furnace efficiency. On the other hand, a large excess of air, while it may give complete combustion, gives poor furnace efficiency because the air will carry a large amount of heat up the stack. It is a matter of great importance that just the right excess of air be admitted to the furnace. Since it is difficult to measure directly the amount of air entering the furnace, an easier method is used. This method consists in analyzing the flue gas to determine the amount of each of its constituents. From this analysis we can easily compute the amount of air entering the furnace. Knowing the composition of flue gas, we can regulate the amount

of air entering the furnace so as to give the proper excess to insure the best economy of operation.

16. Flue Gas Analysis. — There are various types of apparatus on the market for making the analysis of flue gas, most of which are modifications of the apparatus designed by ORSAT. A complete description of the Orsat apparatus will not be given here, but the principle of its operation is as follows.

A sample of gas is taken from the rear of the furnace or between the furnace and the stack. After being cooled to the room temperature, it is carefully measured by volume at atmospheric pressure. All measurements are taken at room temperature and at atmospheric pressure. This known volume of our sample is first passed a few times through a solution of caustic potash, which absorbs the carbon dioxide. The volume is measured again, and the difference between the new volume and the original volume is the volume of the carbon dioxide absorbed. The same sample is next passed several times through a solution of potassium pyrogallate, which absorbs the oxygen. The amount of oxygen is determined by the loss in volume, as before. Next the sample is passed several times through a solution of acid cuprous chloride and the carbon monoxide removed, and its amount determined as before. The amount of carbon monoxide is usually quite small. The remainder of the sample is usually assumed to be nitrogen.

17. Heat Lost in Flue Gas. — The weight of flue gas per pound of fuel burned (assumed carbon and ash) may be computed from the formula,

$$W = 3.032\ \mathrm{C}\left(\frac{\mathrm{N}}{\mathrm{CO_2}+\mathrm{CO}}\right) + (1-\mathrm{A}),$$

where

W = weight of flue gas per pound of fuel burned.

C = decimal part by weight of total carbon in fuel.

N = percentage by volume of nitrogen in flue gas.

CO_2 = percentage by volume of carbon dioxide in flue gas.

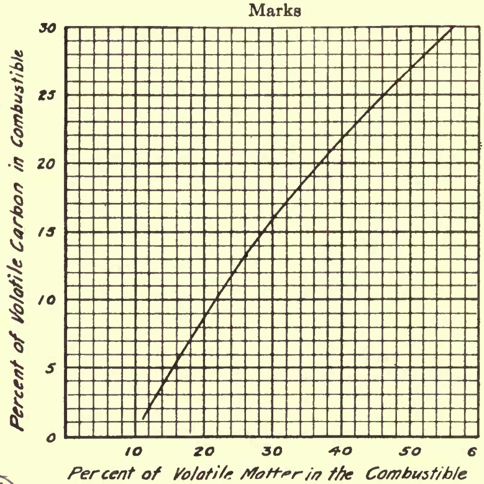
CO = percentage by volume of carbon monoxide in flue gas.

A = decimal part by weight of ash in fuel as fired.

Unless the ultimate analysis of the fuel is known, the weight of carbon in the volatile matter will have to be estimated and added to the weight of fixed carbon to give C in the preceding formula. Marks has published a chart showing the approximate

relation between the volatile carbon in the combustible and the total volatile matter in it. With the aid of this chart (Fig. 4), the value for C in the preceding formula can be approximated from the proximate analysis. The specific heat of the flue gas is usually taken as .24; and the heat lost per pound of fuel burned is equal to the product of the specific heat of flue gas, the weight of gas per pound of fuel, and the difference in temperature between the leaving flue gas and the entering air.

Chart for determining the Carbon in the Volatile Matter



6+65=91
91x13.5=12.1%

FIG. 4

EXAMPLE. How much heat is carried up the stack by the dry flue gases, when the furnace is burning coal of the following proximate analysis? Moisture = 3%, fixed carbon = 65%, volatile matter = 26%, and ash = 6%. The analysis of flue gases gives: $CO_2 = 10\%$, $O = 8\%$, and $CO = .5\%$. The stack temperature is 500° F. and the temperature of the air entering the furnace is 80° F.

SOLUTION. From the chart of Figure 4, we find that the percent of volatile carbon in the combustible is about 13.5, which corresponds to 12.1% of fuel. The total carbon is then 12.1+65 = 77.1%, and the weight of flue gases per pound of coal is

$$3.032 \times .771 \left(\frac{81.5}{10+0.5}\right) + (1 - .03) = 19.1 \text{ pounds.}$$

The heat carried up the stack by the dry flue gases is then $.24 \times 19.1\ (500 - 80) = 1925$ B.t.u. for each pound of coal fired. In this problem, the heat-value of a pound of coal found from the proximate analysis is 13860 B.t.u. Hence the loss is 1925/13860 = 13.9% of the heat available.

Both the fuel and the air contain moisture. This moisture also carries heat up the stack, since it is a superheated vapor upon leaving the furnace.

18. Value of CO_2 for Best Efficiency. — As has been stated, the efficiency of the furnace will vary with the excess of air admitted. Since the percentage of CO_2 also varies with the excess of air, we see that an indication of the efficiency is given by the CO_2 reading. Just what percentage of CO_2 corresponds to the highest operating efficiency depends upon such factors as kind and state of fuel, stack temperature, etc. After the proper amount of CO_2 for best efficiency has been determined under these conditions, the CO_2 reading will indicate whether or not high efficiency is being obtained. Since the determination of the CO_2 is a comparatively simple operation, it is an excellent way to keep check on operating conditions. Some plants go so far as to pay their firemen on the basis of the CO_2 record. In general, high CO_2 means high efficiency, unless there is some abnormal condition such as too much CO. The CO should be kept as near zero as possible.

At the same temperature and pressure, CO_2 occupies the same volume as the oxygen from which it was formed. The volume of the oxygen in the air is 21%. Hence, if the products of combustion are cooled down to the temperature of the entering air, the CO_2 reading would be 21% for perfect combustion with no excess of air, assuming the fuel to be carbon and ash. In practice, the CO_2 runs 17% or lower. Even 15% is usually considered an indication of very good efficiency.

19. CO_2 Recorders. — Automatic devices are on the market that will analyze and record the amount of CO_2 on a chart. An analysis is made every few minutes, so that a complete record is kept of the operating efficiency.

CHAPTER III

STEAM

20. Introduction. Definitions. — A perfect gas may be at any temperature under any pressure. For instance, air may be placed under a certain pressure and have its temperature raised or lowered by the addition or subtraction of heat. On the other hand, a saturated vapor, such as steam, can exist only at a certain definite temperature for each particular pressure. Under ordinary atmospheric pressure, saturated steam can exist only at a temperature of about 212° F. Under an absolute pressure of 100 pounds per square inch, saturated steam will be at a temperature of 327.86° F.

Let us consider a case in which a pound of water at 32° F. is placed under a pressure of, say, 100 pounds per square inch. The containing vessel is supposed to be so constructed that the pressure remains constant, no matter what change of volume takes place. Now suppose that the water is heated. There will be little change in volume, but there will be a rise of temperature of approximately one degree F. for each B.t.u. given to the water. This will continue until we have added 298.5 B.t.u. The temperature will then be 327.86° F. A further addition of heat up to a limit will not cause any change of temperature, but will effect a change in the physical condition of the water, turning it to steam. We shall need to apply 887.6 B.t.u. to effect this change completely. We have now added a total of $298.5+887.6=1186.1$ B.t.u., and have converted the pound of water, originally at 32° F., into ***dry saturated steam*** at a temperature of 327.86° F., and under an absolute pressure of 100 pounds per square inch.

Now that the water is all evaporated, if more heat be added to this steam, the temperature will rise at the rate of nearly two degrees per B.t.u. added. We now have ***superheated steam.***

As stated previously, the volume of the water will change but little until the boiling point is reached. The space occupied by the saturated steam will be 4.432 cubic feet. This is many times greater than the space formerly occupied by the water. Part of the 887.6 B.t.u. that was used to evaporate the water was evidently used to cause this change in volume under the pressure

of 100 pounds per square inch. The remainder was used to make the physical change in the water, to increase the kinetic energy of its atoms. For pressures other than 100 pounds we would have values different from those given above.

Figure 5 represents graphically the relation between the temperature and the heat added to a pound of ice, starting at zero degrees F. (with the assumption that the pressure is constant). Upon the first addition of heat, the temperature of the ice will rise until the melting point is reached. Further addition of heat

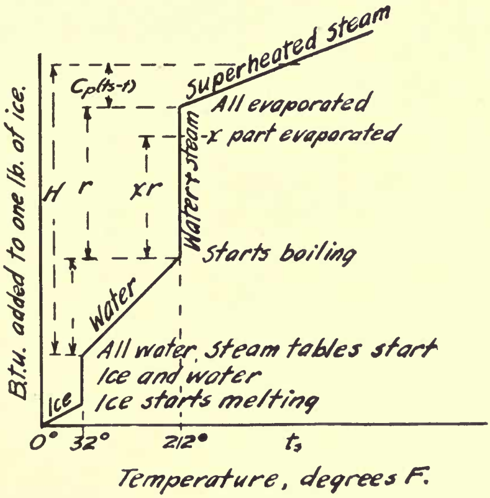


Fig. 5

causes the ice to melt. This occurs without a change of temperature. The part of the line representing the melting of the ice is therefore vertical. When the ice is all melted, addition of more heat causes the temperature of the water to rise. This will continue until the boiling point is reached. That part of the line representing evaporation will be vertical since there is no change of temperature during that period. When evaporation is complete, the addition of heat again causes a rise in temperature.

The amount of heat necessary to raise the temperature of the water, the amount of heat required to change it to steam, and also the volumes of steam formed under different pressures, have been determined by numerous experiments and are published

under the name of ***steam tables.*** We shall use in our work the tables prepared by C. H. Peabody.[1] These are arranged in two ways. In Table I, the various absolute pressures at which water boils are given for each degree F. from 32° to 428°. In Table II, the various temperatures at which water boils are given for each pound per square inch from 1 to 336. The values are arranged in two tables not because they are different, but simply as a convenience in their use.

In Table I, the first column, headed ***t,*** gives the temperature at which water boils. The second column, headed ***p,*** gives the absolute pressure under which it must be in order that it boil at the temperature given in the first column.

The third column, headed ***q,*** gives the ***heat of the liquid,*** *which is the number of B.t.u. necessary to change the temperature of one pound of water from* 32° *F. to the temperature given in the first column.* This does not mean that there is no heat in the water at 32°. The heat in the water below the freezing point is of no moment to the steam engineer; hence it is chosen as the arbitrary starting point.

Column four, headed ***r,*** gives the ***heat of vaporization,*** *which is the B.t.u. necessary to evaporate completely a pound of water at the temperature and pressure given in the first and second columns.* This is sometimes called the ***latent heat of evaporation.*** The sum of the heat of the liquid and the heat of vaporization is called the ***total heat.***

The fifth column, headed ***p,*** *is that part of the heat of vaporization that is used in energizing the atoms of the water to turn it to a vapor;* it is called the ***heat equivalent of internal work.***

The sixth column, headed ***Apu,*** *is the rest of the heat of vaporization, or that part that is needed to do the work of increasing the volume, under the pressure of column two;* it is called the ***heat equivalent of external work.***

Columns seven and eight will not be discussed here. Column nine, headed ***s,*** gives in cubic feet the ***specific volume,*** *which is the volume of one pound of dry saturated steam under the pressure of column two.*

Column ten gives the *reciprocals* of the values found in column nine. It is *the weight of one cubic foot of dry saturated steam under the pressure of column two.*

[1] C. H. PEABODY, *Steam Tables.*

Steam generated in most boilers not equipped with a superheater is likely to carry with it, when leaving through the outlet pipe, a small amount of water in a finely divided state or mist. Steam containing this moisture is said to be ***wet steam.*** The

Chart Showing Specific Heat of Superheated Steam Values from Knoblauch and Jakob

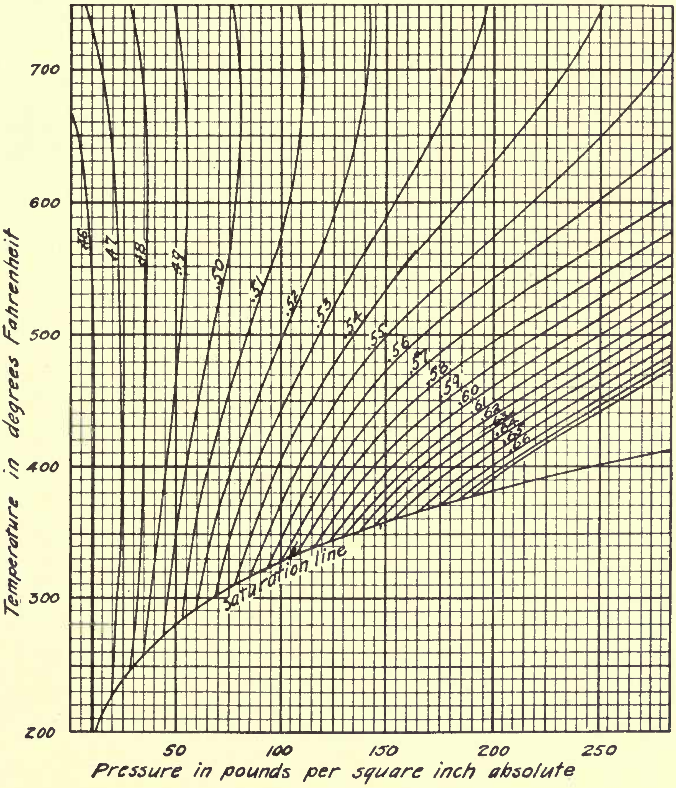


Fig. 6.

quality of wet steam is expressed in percent. If in a hundred parts by weight of a mixture of steam and water, five parts by weight are moisture, the ***quality*** of the mixture is said to be 95% and the ***priming*** 5%.

As long as steam is in contact with water it will remain saturated, and its temperature cannot be raised under constant pres-

sure. If it is conducted away from the water and led to a superheater, its temperature will be raised by the addition of heat. It is then ***superheated steam.*** The amount of heat necessary to superheat depends upon the pressure and upon the degree of superheat. The chart of Fig. 6 gives the specific heat of superheated steam for the ranges of pressure and temperature commonly found in practice. The specific heat of steam varies with both temperature and pressure. The chart gives the average values of specific heat as the steam is raised from the temperature of saturation to the temperature of superheat.

EXAMPLE 1. How much heat is required to change a pound of water at 70° F. into dry saturated steam at a pressure of 120 pounds per square inch absolute?

SOLUTION. On page 48 of Peabody's Steam Tables, we find that the heat of the liquid, q, for 120 pounds pressure is 312.3 B.t.u. This amount of heat would bring the temperature of the water from 32° F. to the boiling point. As the temperature of the water to start with is 70° (p. 36), it already contains 38.1 B.t.u. It is then necessary to add to it $312.3-38.1=274.2$ B.t.u. in order to bring it to the boiling point. To evaporate the water requires the heat of vaporization, r, at 120 pounds (p. 48), which is 876.9. Hence the total heat required to bring the water up to boiling and to evaporate it is $274.2+876.9=1151.1$ B.t.u.

EXAMPLE 2. If, in Example 1, the quality of the steam formed had been 97%, how much heat would it have required?

SOLUTION. The water must *all* be brought to the boiling point, which takes the same amount of heat as in Example 1, 274.2 B.t.u. As the quality is 97%, only $.97\times876.9=850.6$ B.t.u. are needed to evaporate the water. Hence the total amount of heat required is $274.2+850.6=1124.8$ B.t.u.

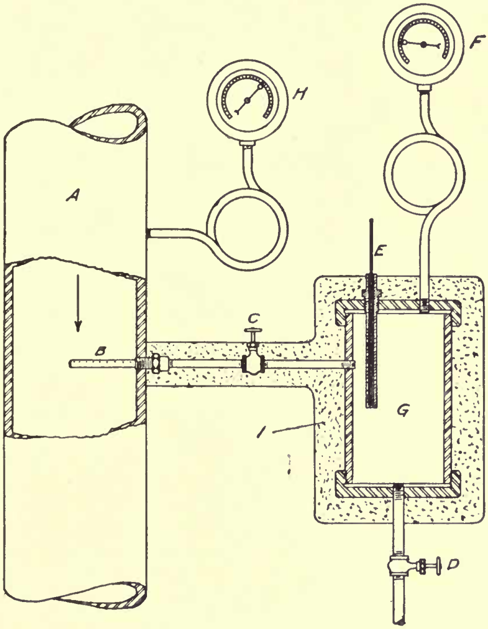
EXAMPLE 3. Find the amount of heat necessary to generate the pound of steam in Example 1, if it is superheated to a temperature of 475° F.

SOLUTION. To generate a pound of dry saturated steam under the conditions of Example 1, requires 1151.1 B.t.u. The temperature corresponding to 120 pounds pressure is 341.3° F. The superheat is then $475°-341.3°=133.7°$. From the chart of Fig. 6, the specific heat of superheated steam is .537. It will take $.537\times133.7=71.8$ B.t.u. to superheat the steam. Hence the total heat required is $1151.1+71.8=1222.9$ B.t.u.

EXAMPLE 4. Find the volume of the pound of steam in Example 2.

SOLUTION. A pound of dry saturated steam at 120 pounds pressure occupies 3.723 cubic feet. The quality being 97%, the volume occupied by the steam is $.97\times3.723=3.611$ cubic feet. A pound of water occupies .016 cubic feet, and, as 3% of the pound of wet steam is water, the volume of the water is $.03\times.016=.0005$ cubic feet. Hence the total volume is $3.611+.0005=3.611$ cubic feet.

21. The Steam Calorimeter. — In making tests of boilers or engines it is necessary to know the quality of steam leaving the one or entering the other. A ***steam calorimeter*** is used in making this determination. Several types of calorimeters are in use. If the quality of steam is high (between 94% and 100%), the throttling type is usually used. When properly constructed this calorimeter is sufficiently accurate for ordinary purposes.



Throttling Calorimeter

Fig. 7

Figure 7 shows a ***throttling calorimeter*** attached to a steam-pipe A. If the steam is saturated, and the pressure is known from a pressure gage H, its temperature may be determined from the steam tables. If the steam in A is superheated, it is also necessary to take its temperature by means of a thermometer, and the heat contents may be calculated from the steam tables. If it is wet, the quality must be known to find its heat contents. The tube B in Fig. 7 is a sampling tube through which a sample of steam is taken from the pipe A. This sample is throttled down

in pressure at C from the pressure in A, say p_1, to the pressure in the calorimeter G, say p_2. If the calorimeter is well covered, but little heat is lost by radiation and the heat contents in one pound of steam in A is the same as in the chamber G. The pressure and temperature in G are measured by the gage F and the thermometer E, and the heat contents are computed by means of the steam tables.

Since the heat content is the same per pound in A as in G, the quality in the former may be computed as follows. The total heat in A equals q_1+xr_1 where x is the quality and q_1 and r_1 are the heat of liquid and the heat of vaporization in A, respectively. If the calorimeter is working properly, the steam will be superheated in G, and its heat contents will be equal to

$$q_2+r_2+.48\ (t_3-t_2),$$

where q_2 and r_2 are the heat of the liquid and the heat of vaporization in G, respectively, t_3 is the temperature of steam in G as measured by the thermometer E, and t_2 is the temperature of saturated steam at the pressure p_2. The term $.48\ (t_3-t_2)$ is the heat of superheat in G, since .48 is the specific heat of superheated steam at low pressure and temperature. Then we have

$$q_1+xr_1=q_2+r_2+.48\ (t_3-t_2),$$

from which x may be found.

EXAMPLE. Find the quality of steam leaving a boiler when the pressure is 165 pounds gage. The gage pressure in the throttling calorimeter is 3 pounds, the temperature is 265° F., and the barometer reading is 29.6 inches.

SOLUTION. From the steam tables, $q_1=345$, $r_1=851$, $q_2=189$, $r_2=964$, and $t_2=221$. Then

$$345+x851=189+964+.48(265-221),$$

from which $x=.975$ or 97.5%.

CHAPTER IV

BOILERS

22. Introduction. — The stored energy in fuels is utilized by means of the steam engine as follows. The fuel is burned in a furnace, resulting in a mixture of heated gases. These hot gases pass over and along the surface of a boiler. A transfer of heat takes place through those boiler surfaces that are exposed to contact with the hot gases or to radiation from the incandescent fuel bed on the one side and water or steam on the other side. This boiler surface is called ***heating surface.*** This heat is conducted through the shell of the boiler and is spent in heating and evaporating the water contained in the boiler. The steam thus formed is conducted from the boiler to the engine or turbine, where it does work due to its pressure and to its tendency to expand.

Practically all boilers have a considerable storage of heated water and steam. This water and steam is under a high pressure and would increase in volume hundreds of times if the pressure were removed. A sudden release of this pressure causes an explosion. Many lives have been lost and a great amount of property destroyed by boiler explosions. Hence the consideration of first importance in a boiler is its safety. Other considerations are its first cost, its life, and the ease with which it may be cleaned and repaired. In portable boilers and marine boilers, weight and the space occupied are of great importance.

Since the purpose of the heating surface is to conduct heat from the furnace to the water, it follows that the conduction should be rapid and effective. To be efficient, the boiler must extract a large proportion of the heat generated in the furnace. The surface must be of such size and so arranged that time is allowed to render this transfer as complete as possible.

Due to the erosion of some of the parts, or due to overheating consequent on the formation of scale, a boiler originally fit for a certain class of service may become so weakened that it is unsafe for high pressures. Many states provide for an inspection of boilers and equipment in order to prevent explosions. The various boiler insurance companies also make periodic inspection of insured boilers. As a result of this inspection, the inspector sets

a limit to the pressure that the boiler may carry. The fireman must be constantly on the alert to detect any signs of a developing weakness.

Much of the water used in boilers will deposit scale on the heating surface of the boiler. This scale greatly hinders the conduction of heat to the water and may even become thick enough to allow the metal on which it settles to become overheated to such an extent that it gives way and causes an explosion. If the scale becomes too thick, it must be removed. The removal and prevention of scale will be considered later.

The fire side of the heating surface in many boilers will collect soot and fine ash. To maintain efficient steaming these surfaces must be kept clean. The soot should be swept or blown from the surface as fast as it collects.

In the construction of a steam-boiler, the following requirements are to be considered.

(1) Proper materials of uniform strength and reliability must be employed, and the size of all parts must be so designed that a sufficient factor of safety exists. The workmanship must be good.

(2) There must be steam space and water capacity such that a sudden change of load will not cause an undue drop in steam pressure.

(3) There must be a sufficient water surface to allow for complete separation of the steam from the water. Too small a surface will result in foaming.

(4) A thorough circulation of the water must be provided, so that a uniform temperature is maintained throughout the boiler. Water is a very poor conductor of heat, and it is therefore essential that there be a continuous flow over the heating surface.

(5) Stresses due to temperature change must be eliminated.

(6) In so far as possible, all joints or seams must be protected from the direct action of the flame.

(7) Access must be possible to all parts for cleaning and repair.

(8) A means for the discharge of mud or sludge that is left by the evaporation of the water must be provided.

The modern steam boiler is the result of an evolution starting with a vessel much resembling a closed kettle. The pressure in the early boilers was nearly atmospheric; hence the shape was not influenced by the consideration of the strength of the boiler.

With the use of steam under pressure, boilers assumed a cylindrical or spherical shape, since these shapes are not distorted so much by internal pressure. The simplest boiler is cylindrical, with hemispherical ends.

For a given steam pressure, the thickness of metal required varies directly as the diameter. Hence heavy plate must be used for boilers of considerable size. Moreover, the ratio between the heating surface and the volume decreases as the diameter increases. Thus it is seen that the single cylinder is suitable for small boilers only.

From this early and simple type of boiler the development has proceeded along two distinct lines. First, in place of a single large cylinder, several smaller ones were sometimes used, thereby decreasing the weight of metal and increasing the rate of steaming. Carrying this idea still further results in a large number of very small cylinders or tubes filled with water and surrounded by fire. This is the modern ***water-tube boiler.***

The other way of increasing the heating surface of a cylindrical boiler is to run smoke flues through it. The earliest types contain one or two large flues. The modern type contains a large number of small tubes through which the fire or the products of combustion pass. This type is known as the ***fire-tube boiler.***

23. Rated Horsepower. — The rating of a boiler is usually based on its heating surface. There is no standard for this rating. It is becoming general practice, however, to rate a water-tube boiler on a basis of 10 *square feet of heating surface per boiler horsepower*, and to rate a fire-tube boiler at 11 *or* 12 *square feet per boiler horsepower.* Under average conditions of firing, care, and draft, a boiler should develop good economy at its rated horsepower. However, many boilers are able to carry great overload and still give excellent efficiency. Poor efficiency is due in general more to overloading the furnace than to increasing the evaporation from the boiler.

24. Heating Surface. — It has been customary to consider as heating surfaces those surfaces which have water on one side and the products of combustion on the other side. No distinction is made in the thickness of metal in different parts of the boiler, or in the difference in temperature of the gases on their path along the heating surface. However, great accuracy is seldom required

in computing the heating surface of a boiler. In calculating heating surface, the inside of fire tubes and the outside of water tubes is used.

25. Rules for Finding the Heating Surface. —

1. *Horizontal Return-tubular boilers.* Kent [1] gives the following rule:

Take the dimensions in inches. Multiply two-thirds of the circumference of the shell by its length; multiply the sum of the circumferences of all the tubes by their common length; to the sum of these products add two-thirds the area of both tube sheets; from this sum subtract twice the area of all the tubes; divide the remainder by 144 to obtain the area in square feet.

2. *Vertical Tubular boilers.* Kent [2] gives the following rule:

Multiply the circumference of the fire-box (in inches) by its height above the grate; multiply the combined circumference of all the tubes by their length, and to these two products add the area of the lower tube sheet; from this sum subtract the area of all the tubes, and divide by 144: the quotient is the number of square feet of heating surface.

3. *General rule.* The U. S. Bureau of Mines [3] gives the following rule:

A short approximate method for any boiler is to figure the square feet of heating surface in the tubes and divide it by 0.85 for a return tubular boiler or by 0.95 for a water tube boiler. In case the return tubular boiler has an arch over the top for gas passage, giving the so-called third return, it is necessary to add from 100 to 200 square feet to the result to obtain the total heating surface.

In this last rule the heating surface in fire-tube boilers is figured from the outside diameter of tubes.

26. Superheating Surface. — In modern practice, steam is often led off from the main steam space and taken through other heating coils. Since there is no water in contact with this steam, it

[1] Kent, *Mechanical Engineers' Handbook*, 1916 Edition, p. 888.

[2] *Ibid.*

[3] Bureau of Mines, U. S. Department of the Interior, *Bulletin No.* 40, p. 9.

will be superheated. That surface which has this superheated steam on one side and fire or hot gases on the other is called the ***superheating surface.***

27. Size of Boiler Tubes. — The outside diameter is the nominal diameter in boiler tubes. The following table given by the

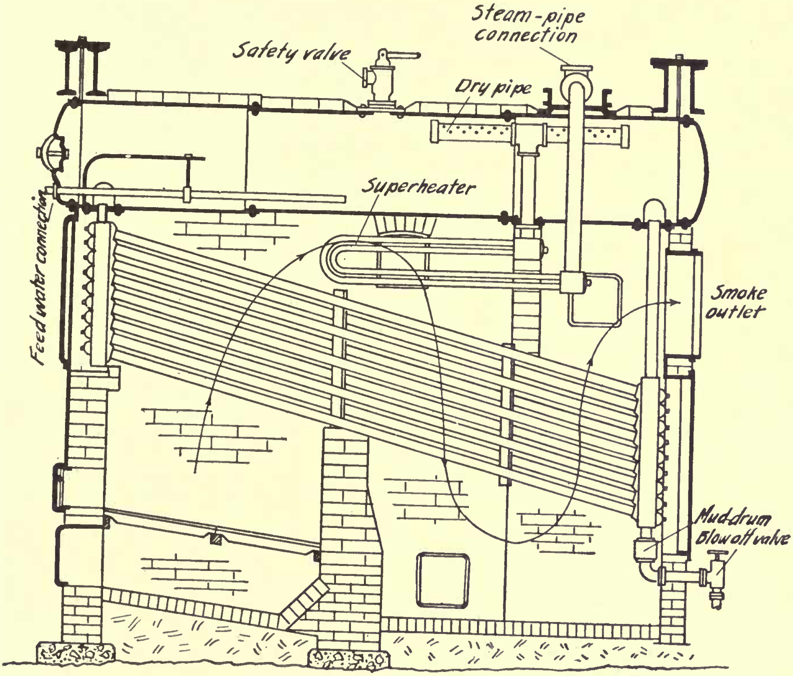


FIG. 8

National Tube Works gives the nominal or outside diameter, the inside diameter, and the thickness, of standard lap-welded boiler tubes.

SIZE IN INCHES OF STANDARD LAP-WELDED TUBES

External Diameter......	1.0	1.25	1.5	1.75	2.0	2.25	2.5	2.75	3.0
Internal Diameter......	.810	1.060	1.310	1.560	1.810	2.060	2.282	2.532	2.782
Standard Thickness.....	.095	.095	.095	.095	.095	.095	.109	.109	.109
External Diameter......	3.25	3.5	3.75	4.0	4.5	5.0	6.0	7.0	8.0
Internal Diameter......	3.010	3.260	3.510	3.732	4.232	4.704	5.670	6.670	7.675
Standard Thickness.....	.120	.120	.120	.134	.134	.148	.165	.165	.165

28. Water-tube Boiler. Babcock and Wilcox Type. — One of the most common forms of water-tube boilers is the Babcock and Wilcox type, shown in Fig. 8. Here the tubes are fastened into headers, which in turn are connected by other tubes to a forged steel cross-box that is riveted to the steam drum above, as shown in Fig. 9. Headers are made of cast-iron for low-pressure work, and of wrought steel for high-pressure work. They are of two types, vertical and inclined. In the latter the tubes enter the header at right angles. The tubes between the front and rear headers are inclined so that a circulation of water is insured. Opposite the end of each tube in the header is placed a hand-hole to permit cleaning and access to the tube. These holes are closed by means of a cap. This arrangement also allows ease in cleaning the scale from the tubes, as the caps are easily removed and a cleaner run through the tube. Clean-out doors are placed in the setting opposite the headers to give easy access for cleaning.

The grates are so located, and fire-brick partitions or baffles are so placed, that the hot gases usually pass across the tubes three times on their way to the stack. The mud drum is connected to the lower end of the back header. This collects the sediment that is formed during the evaporation of the water. This sediment is blown off from time to time through the blow-off pipe shown at the lower right hand of Fig. 8.

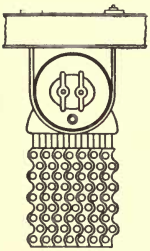


FIG. 9

The feedwater is brought in at the front of the steam drum, and is so delivered that its velocity will aid in circulation of the water in the boiler. The circulation is back through the steam drum, down the tubes to the back header, where it is distributed to the inclined tubes, up these to the front header, and thence to the front of the steam drum. The steam is taken off through a dry-pipe located at the top of the steam drum. If a superheater is attached, as shown in Fig. 8, the steam is led from the dry-pipe at the top of the steam drum into and through the superheater, and then up to the steam pipe shown at the top of the steam drum. Provision is made for flooding the superheater during the period of getting up steam. This keeps the surface cool enough to prevent its burning out. The boiler is carried by slings from horizontal girders placed above it. The whole boiler is surrounded by a smoke-tight brick setting.

A similar form, but using boiler plate headers instead of cast or forged sectional headers, is called the Heine type.

29. Water-tube Boiler. Stirling Type. — A common form of water-tube boiler is shown in Fig. 10. Here banks of tubes lead

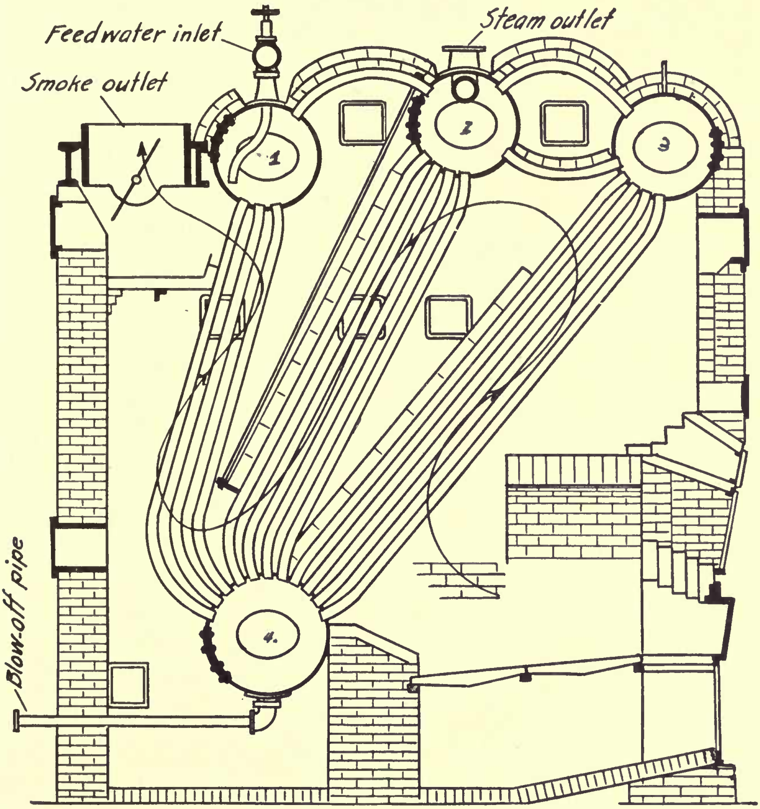


FIG. 10

from the lower drum, or mud drum, to the three drums above. The water is taken in at the rear upper drum and the steam is drawn off at the top of the middle drum. Arched tubes connect the steam space of the upper drums. Access to the inside of the drums, for the purpose of expanding the tubes, for cleaning, and

for inspection, is had by means of a manhole located at the end of each drum. The sediment collects and is blown off at the bottom of the mud drum through the blow-off pipe and valve shown at the lower left hand corner of Fig. 10. The baffling is

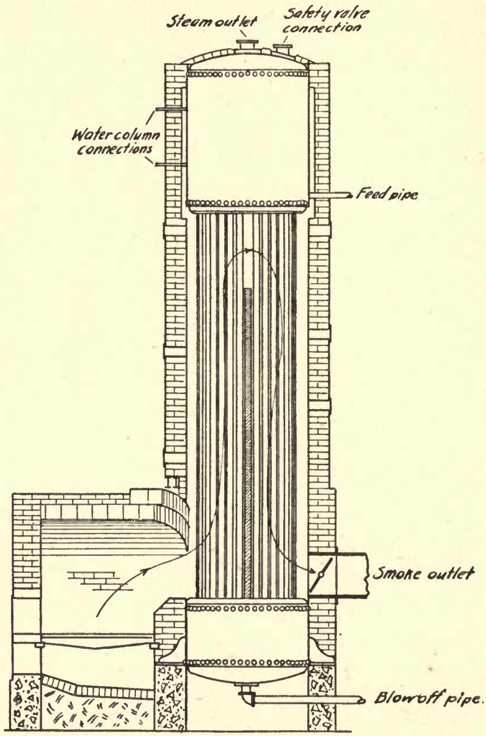


Fig. 11

so arranged that the products of combustion are forced to travel practically the entire length of all the banks of tubes, up the first bank, down the second and up the third. Clean-out doors for the removal of soot are located at various points shown in Fig. 10.

30. Water-tube Boiler. Vertical Type. — A Wickes vertical water-tube boiler is shown in Fig. 11. In this type of boiler the tubes are straight and are placed vertically. They are connected to a mud drum below and to a steam drum above. The products

of combustion pass up along the front half of the tubes and down the back half, leaving at the rear.

31. Fire-tube Boiler. Externally Fired, Return-tubular Type. — In this country, where moderate pressures are wanted, the common type of fire-tube boiler is the externally fired return-tubular boiler. Figure 12 shows the construction of a boiler of this type. It consists of a cylindrical shell, made of steel or wrought-iron plates rolled into a cylindrical shape and riveted together. The ends or heads are formed from flat circular plates flanged around the outer edge and riveted to the cylindrical shell. A large number of fire-tubes extend from one end-sheet to the other. They occupy the lower two-thirds of the shell, the top third being steam space. The flat plates or tube-sheets tend to bulge outward with the internal pressure. To prevent this, the part of the sheets above and below the level of the tubes must be stayed. These stays are of two kinds, "through stays," and "diagonal stays." The former are steel rods that run the entire length of the shell, pierce the tube-sheets and hold the sheets in by means of nuts. The latter run from each tube-sheet diagonally back to the shell. In certain types of large boilers, some of the tubes are made heavier and are threaded on the ends. These tubes are secured to the end-sheets by means of nuts on the outside. Such tubes are called *stay-tubes*. In general, the tubes act as stays, and that part of the ends occupied by the tubes needs little if any extra staying. The tubes are expanded into the sheet and their ends are beaded over.

The feed-pipe enters the front end of the boiler just below the water line, and the steam leaves by the dry-pipe, which leads out of the top of the shell. The mud is blown off through an outlet at the bottom near the rear. If the feedwater is taken from a pond or stream and contains much vegetable matter, there should be a surface blow-off to skim the scum that forms on top of the water. A manhole is located at the top near the rear and a hand-hole in front beneath the tubes. The grate is put beneath the front end of the shell. The products of combustion pass over the bridge wall, back along the bottom of the shell, enter the tubes from the rear, pass through them and out of the uptake at the front of the boiler or through a flue over the top of the boiler to an uptake at the rear.

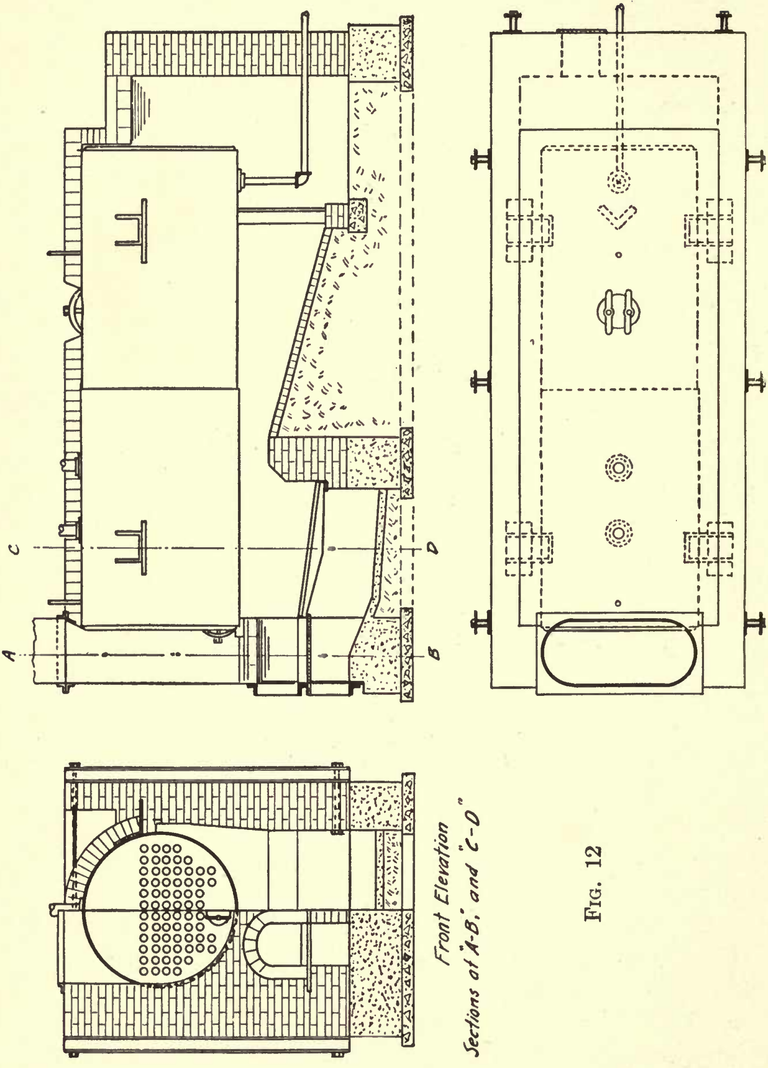


Fig. 12

The shell is supported by brackets which are riveted to the shell and rest on the brickwork of the setting. The rear bearing is fitted with rollers to allow the boiler to expand without cracking the setting.

32. Fire-tube Boiler. Internally Fired, Return-tubular Type. — Instead of having the fire outside the shell, as in the preceding boiler, the fire is sometimes placed inside one or more large

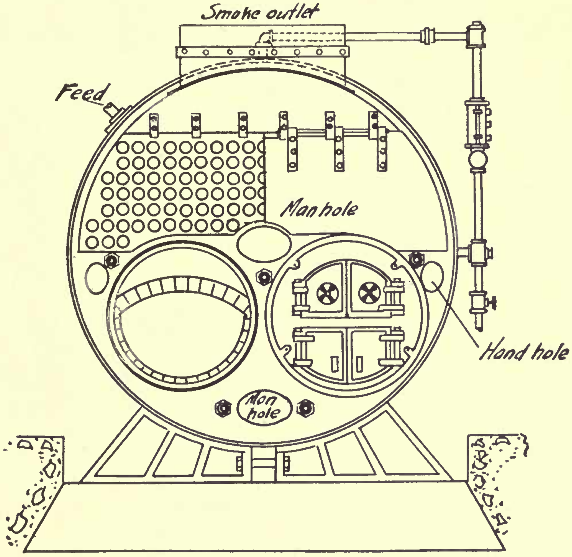


FIG. 13a

fire-flues running the whole length of boiler. Figure 13 shows a Springfield boiler of this type. These large fire-flues are of course subjected to external pressure, and therefore have a tendency to collapse. In order to prevent this, they must be stiffened or stayed in some way. This is ordinarily done by rolling them in corrugations. If the flue is braced in such a manner that the metal at any one point is very thick, there is danger of its being overheated, since the gases are hottest in this combustion flue.

The fire is at the front of the flue. The gases pass back through the flue to the rear of the boiler, into the combustion chamber. The heating surface is mostly composed of the comparatively thin flue and tubes. The thick outer shell is not subjected to

the high temperatures of the combustion chamber, as in the externally fired boiler.

33. Fire-tube Boiler. Scotch Marine Type. — While water-tube boilers are used to some extent in marine service, the standard boiler is the Scotch marine type. This is very similar to the internally fired boiler described in § 32, which is also sometimes called a Scotch marine boiler. It commonly has three combus-

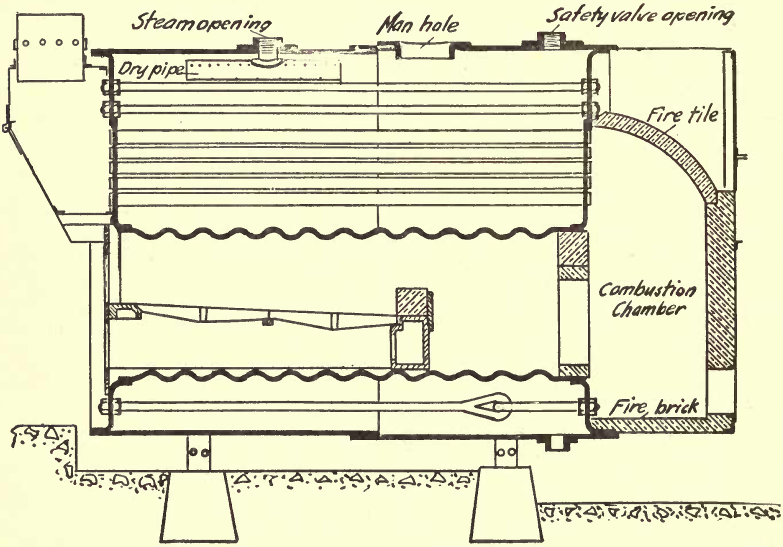


Fig. 13b

tion flues, each of which has its own set of tubes. In marine practice the combustion chamber at the rear of the flue and the tubes is internal, there being a water-leg between the combustion chamber and the rear head. There is a combustion chamber for each flue and its set of tubes. Since the surfaces of the combustion chamber are flat, they must be stayed.

These boilers are large in diameter; the outer shell must therefore be very thick. The longitudinal seams are usually triple riveted, with two-strap butt joints. The Scotch marine boiler is used to some extent on land, but as the space occupied is usually an element of less importance here, the type is not common.

34. Fire-tube Boiler. Vertical Type. — A boiler often used for small or portable plants is the vertical fire-tube type (Fig. 14). These boilers are internally fired, the fire being enclosed in the lower part of the shell, and surrounded by an annular water-ring or water-leg. The lower tube-sheet is placed but a small distance above the grate. Therefore the space for combustion is very limited. The tubes are vertical and are expanded into the lower and the upper tube-sheets.

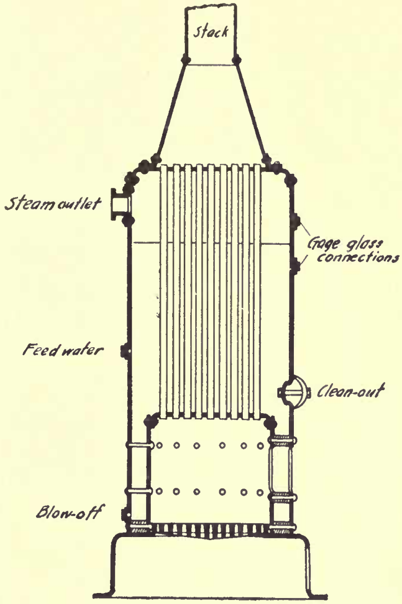


Fig. 14

35. Fire-tube Boiler. Locomotive Type. — The type of boiler used on locomotives, and also often used on portable plants, is shown in Fig. 15. In this boiler, the fire-box is at the rear end of the shell, and its top and sides are water-heating surfaces.

Since the sheets that form the water-legs at the sides and rear of the fire-box are flat, it is necessary to stay them to prevent distortion. A screw-stay is used for this purpose; it consists of a threaded bolt screwed through the parallel plates. The threads on the center part of these stay-bolts are removed so that cracks will not start in the bolt at the root of the thread. On what are called *safety* stays a small hole is drilled in from the end so that a cracked bolt will leak steam and give warning.

The flat or arched sheet at the top of the fire-box is called the *crown-sheet.* The crown-sheet is stayed in various ways, sometimes by radial stay-bolts which run between it and the outer shell, or by sling stays, which are girders slung from the outer shell. Fire-tubes extend from the tube-sheet at the front of the fire-box to the tube-sheet at the front end of the boiler.

The tubes in locomotive boilers are smaller than in the types previously described, and are placed as close together as good

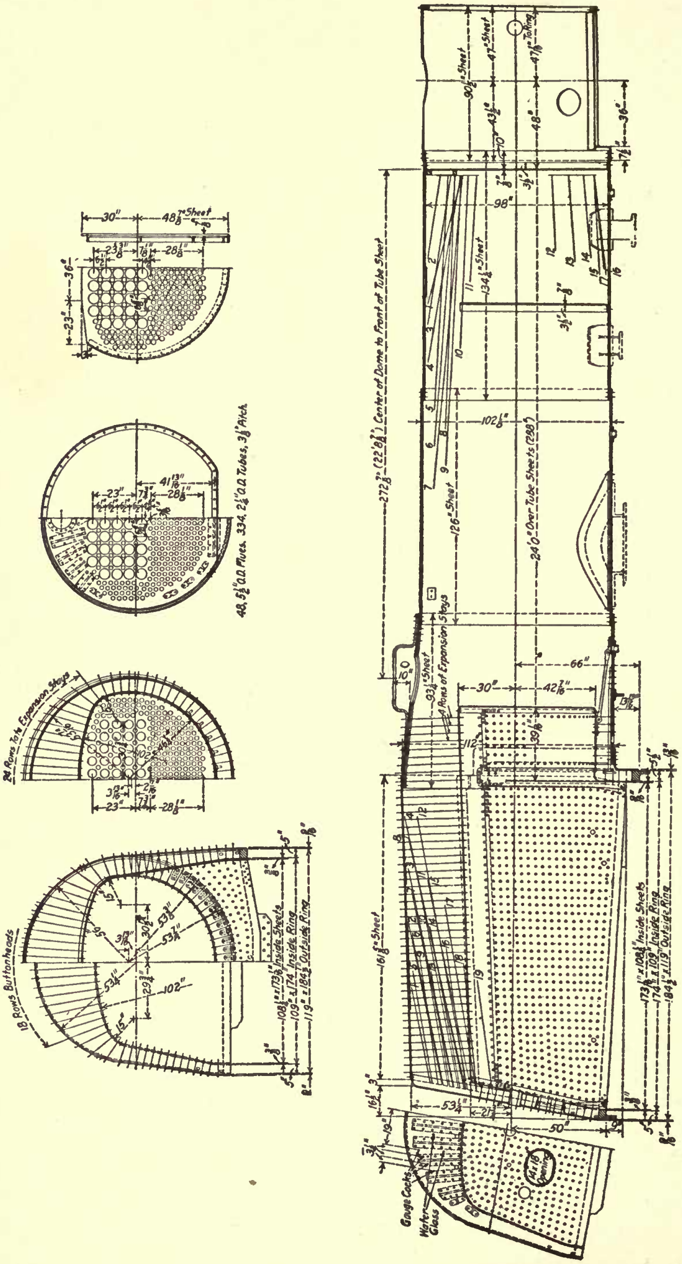


FIG. 15.—Longitudinal and Cross Sections of Boiler of Articulated Compound Locomotive

circulation of the water will permit. By making the tubes small and numerous, a large heating surface is obtained. Where a superheater is used, as shown in Fig. 15, some of the tubes are made larger and contain the superheating surface. This superheating surface is formed by tubes that extend into the fire-tubes from the front end of the boiler and run to within a short distance of the fire-box.

The outer shell extends beyond the front tube-plate to form a smoke-box. In this smoke-box vertical nozzles are located, through which the exhaust steam from the engines escapes. This induces a strong draft that allows a very rapid rate of combustion. The rate of combustion often exceeds 100 pounds of coal per square foot of grate surface per hour.

Since the steaming of this type of boiler is very rapid, the steam is taken from a steam-dome located on the top of the shell. This allows the steam to be taken at a distance from the water surface, thereby insuring fairly dry steam. The throttle valve for the engine is located in the dome.

In smaller locomotive boilers, the fire-box is set between the rear drivers. In the larger sizes, this arrangement will not allow a large enough grate area, and so the fire-box is extended laterally over low trailer wheels. Manholes and hand-holes are employed to give access for cleaning, as in other types of fire-tube boilers.

36. Superheaters. — During the past few years superheated steam has come into quite general use, especially if it is to be used in steam turbines. The amount of superheat used is generally not large, usually between 100° and 200° Fahrenheit. The advantages gained by the use of superheated over saturated steam will be considered later.

There are two types of superheater. One type is independently fired. The other is formed by the addition of some superheating surface to the main boiler. The latter is the more common form. In this type the steam is taken from the steam space and led through superheating coils. Often provision is made for the flooding of these coils during the period in which steam is being raised, in order to prevent the coils from burning out. The boiler shown in Fig. 8 has a common form of superheater attached.

37. Horsepower of Boilers.—As explained previously, boilers are generally rated by the manufacturer on the amount of their heating surface. The rate at which a boiler is working, however, must be determined from a consideration of the amount of steam that is being generated.

The amount of heat necessary to evaporate a given quantity of water varies with the temperature of the feedwater, with the pressure at which the steam is formed, and with the quality of the steam produced. Hence it is desirable that there be a standard of temperature and pressure at which we can find the equivalent amount of water evaporated, using the same amount of heat as is used under the *actual* conditions of temperature and pressure. The conditions of temperature and pressure set by the A. S. M. E.[1] as a standard are 212° F. and 14.7 pounds per square inch absolute. The *equivalent evaporation* is then the amount of water that would be evaporated from and at 212° F. if the same amount of heat were used in its evaporation as is used in the evaporation under the actual working conditions. From the steam tables, the B.t.u. required to evaporate one pound of water from and at 212° F. is seen to be 969.9 B.t.u. This is the unit of evaporation.

At the time when it first became necessary to rate boilers, a good engine used about 30 pounds of steam per horsepower per hour at a pressure of 70 pounds gage. The judges at the Centennial Exposition in 1876 awarded prizes using the following unit as a standard. *A one-horsepower boiler is one that will evaporate* 30 *pounds of water per hour from feedwater at* 100° *F. into steam at* 70 *pounds pressure by gage.*

The A. S. M. E. has since adopted an equivalent standard and defines a **boiler horsepower** to be *the evaporation of* 34.5 *pounds of water per hour at* 212° *F. into steam at* 212° *F. and at a pressure of* 14.7 *pounds absolute pressure.* As the heat of vaporization at 212° F. or 14.7 pounds absolute is 969.7 B.t.u., it therefore takes 969.7×34.5 B.t.u. per hour for one boiler horsepower.

To determine the horsepower at which a boiler is working, we must therefore first find how many B.t.u. are used to generate one pound of steam under the given conditions of steam pressure and temperature of feedwater. The number of pounds of water evaporated per hour multiplied by the number of B.t.u. to gen-

[1]A. S. M. E. POWER TEST CODE, Edition of 1915, Table 4, §21.

erate one pound of steam under the conditions will give the number of B.t.u. used by the boiler per hour. This product divided by 969.7×34.5 will give us the horsepower of the boiler.

EXAMPLE. It is required to find the horsepower of a boiler working under the following conditions:

Steam pressure = 115 pounds gage.
Temperature of feedwater = 65° F.
Quality of steam = 98% (*i.e.*, 2% priming).
Water fed to boiler per hour = 3640 pounds.

SOLUTION. From the steam tables it is seen that
The heat of the liquid at 115 lb. gage (129.7 lb. abs.) = 318.4 B.t.u.
The heat of vaporization at 115 pounds gage = 872.3
The heat of the liquid at 65° F. = 33.1 B.t.u. The heat required to evaporate one pound of water under the above conditions is $318.4 + .98 \times 872.3 - 33.1 = 1140$ B.t.u., and the B.t.u. used per hour is $1140 \times 3640 = 4150000$. Hence the horsepower of the boiler is $4150000/(34.5 \times 969.7) = 124$ h.p.

38. Factor of Evaporation. — Since it takes 969.7 B.t.u. to evaporate a pound of water from and at 212° F., and since it takes more (1140 B.t.u. in the previous example) to evaporate a pound of water under the actual conditions that exist in the boiler, a certain ratio exists between these amounts. ***The factor of evaporation*** *is the ratio of the amount of heat required to evaporate a pound of water under actual conditions to the amount required to evaporate a pound from and at* 212° *F.* In the previous example, the factor of evaporation was $1140/969.7 = 1.176$. If the factor of evaporation is known, the equivalent evaporation is found by multiplying the actual evaporation by this factor.[1]

By this method, the heat used to raise the temperature of the moisture in the steam from the temperature of the feed water to that of the steam is not considered in computing the factor of evaporation. In most cases the difference in results due to this omission is very small.

39. Efficiency of Boilers. — Usually speaking, the efficiency of anything is the ratio of what is got out to what is put in, output and input being measured in like units. For boilers, *the term* ***efficiency*** *means the ratio of the number of B.t.u. in the steam generated to the number of B.t.u. available in the coal fired.* Boiler efficiency is usually expressed in percent.

[1] In the A. S. M. E. POWER TEST CODE the "Mean B.t.u." and steam tables by MARKS and DAVIS are used, thereby giving 970.4 B.t.u. instead of 969.7 B.t.u. as used above for heat required to evaporate a pound of water from and at 212° F. See POWER TEST CODE, Edition of 1915, pp. 28 and 47.

The fact that the combined efficiency of a boiler, furnace, and grate is not 100% is due to several losses. These losses are due to the following causes.

(1) A part of the fuel may drop through the grate and be lost in the ash.

(2) Heat is lost up the stack. There are several sources of this loss, and to them is due the greatest loss in efficiency. First, unburned particles of solid fuel are often carried from the furnace. The amount depends upon the draft and the kind of fuel. In locomotives, with their high draft and with a fine fuel, this loss may be considerable. Second, there is loss due to the unburned or partially burned hydrocarbons. Black smoke is caused by the incomplete burning of some of the hydrocarbons. Third, heat is carried away by the excess air and the inert nitrogen which have been heated, and by the hot products of combustion. Fourth, heat is required to evaporate and to superheat the moisture in the fuel and in the air. Fifth, there may be loss due to the burning of the carbon to carbon monoxide instead of to carbon dioxide.

(3) Heat is lost by radiation from the furnace and from the boiler.

It is very difficult to separate all these losses and the attempt is seldom made. It must be remembered that what is often called boiler efficiency is really the combined efficiency of grate, furnace, and boiler.

Under the most favorable conditions, using coal as a fuel, efficiencies of over 80% have been attained. Under ordinary conditions of operation, efficiencies vary from 80% to less than 50%. The efficiency may sometimes exceed 80% when underfeed stokers, described later, are used. When oil is used as a fuel, higher efficiencies may be attained, due in part to the better mixing of the air and the fuel.

Example. It is required to find the combined efficiency of a boiler, furnace, and grate, working under the following conditions:

Steam pressure = 127 pounds gage.
Superheat = 190° F.
Temperature of feedwater = 180° F.
Water fed to boiler per hour = 8750 pounds.
Coal fired per hour = 1160 pounds.
B.t.u. per pound of coal as fired = 11540 B.t.u.

Solution. The B.t.u. required to generate one pound of steam under the above conditions is seen to be

$$325.4+866.8-148+.55\times190=1148.7 \text{ B.t.u.}$$

The total B.t.u. used in the generation of steam per hour then is $8750\times1148.7=10051000$ B.t.u. The total B.t.u. in coal fired per hour is $1160\times11540=13386000$ B.t.u. Hence the efficiency is $10051000/13386000=.752$ or 75.2%.

40. A. S. M. E. Boiler Test Code.[1] — In reporting the results of a steam-boiler test it is well to put them in the form prescribed by the A. S. M. E. This form is as follows.

DATA AND RESULTS OF EVAPORATIVE TEST
CODE OF 1915

(1) Test of........................boiler located........................
To determine..
Test conducted by...

Dimensions

(2) Number and kind of boilers..
(3) Kind of furnace...
(4) Grate surface (width........length........)................sq. ft.
(a) Approximate width of air openings in grate.................in.
(b) Percentage of area of air openings to grate surface.......per cent
(5) Water heating surface..sq. ft.
(6) Superheating surface...sq. ft.
(7) Total heating surface..sq. ft.
(a) Ratio of water heating surface to grate surface................
(b) Ratio of total heating surface to grate surface................
(c) Ratio of minimum draft area to grate surface...................
(d) Volume of combustion space between grate and heating surface ....cu. ft.
(e) Distance from center of grate to nearest heating surface.......ft.

Date, Duration, etc.

(8) Date..
(9) Duration..hr.
(10) Kind and size of coal..

Average Pressures, Temperatures, etc.

(11) Steam pressure by gage...........................lb. per sq. in.
(a) Barometric pressure........................in. of mercury
(12) Temperature of steam, if superheated..........................deg.
(a) Normal temperature of saturated steam..................deg.
(13) Temperature of feedwater entering boiler.......................deg.
(a) Temperature of feedwater entering economizer.............deg.
(b) Increase of temperature of water due to economizer.........deg.

[1] A. S. M. E. Power Test Code, Edition of 1915, p. 51.

(14) Temperature of escaping gases leaving boiler....................deg.
 (*a*) Temperature of gases leaving economizer.................deg.
 (*b*) Decrease of temperature of gases due to economizer.........deg.
 (*c*) Temperature of furnace...................................deg.
(15) Force of draft between damper and boiler................in. of water
 (*a*) Draft in main flue near boiler.......................in. of water
 (*b*) Draft in flue between economizer and chimney.......in. of water
 (*c*) Draft in furnace...................................in. of water
 (*d*) Draft or blast in ash-pit...........................in. of water
(16) State of weather..
 (*a*) Temperature of external air..............................deg.
 (*b*) Temperature of air entering ash-pit......................deg.
 (*c*) Relative humidity of air entering ash-pit...............per cent

Quality of Steam

(17) Percentage of moisture in steam or number of degrees of superheating ....per cent or deg.
(18) Factor of correction for quality of steam..........................

Total Quantities

(19) Total weight of coal as fired...................................lb.
(20) Percentage of moisture in coal as fired.....................per cent
(21) Total weight of dry coal (Item $19 \times \left(\frac{1-\text{Item } 20}{100}\right)$)...............lb.
(22) Ash, clinkers, and refuse (dry)
 (*a*) Withdrawn from furnace and ash-pit........................lb.
 (*b*) Withdrawn from tubes, flues and combustion chamber.........lb.
 (*c*) Blown away with gases.....................................lb.
 (*d*) Total...lb.
 (*e*) Weight of clinkers contained in total ash.................lb.
(23) Total combustible burned (Item 21 − Item 22*d*)...................lb.
(24) Percentage of ash and refuse based on dry coal..............per cent
(25) Total weight of water fed to boiler.............................lb.
(26) Total water evaporated, corrected for quality of steam (Item 25 × Item 18)...lb.
(27) Factor of evaporation based on temperature of water entering boiler....
(28) Total equivalent evaporation from and at 212 degrees (Item 26 × Item 27)...lb.

Hourly Quantities and Rates

(29) Dry coal per hour...lb.
(30) Dry coal per square foot of grate surface per hour..............lb.
(31) Water evaporated per hour, corrected for quality of steam.......lb.
(32) Equivalent evaporation per hour from and at 212°................lb.
(33) Equivalent evaporation per hour from and at 212° and per square foot of water heating surface.......................................lb.

Capacity

(34) Evaporation per hour from and at 212° (Same as Item 32)........lb.
 (*a*) Boiler horsepower developed (Item 34 ÷ 34½)...........bl.-h.p.

(35) Rated capacity per hour, from and at 212°........................lb.
(a) Rated boiler horsepower..............................bl.-h.p.
(36) Percentage of rated capacity developed.......................per cent

Economy

(37) Water fed per pound of coal as fired (Item 25÷Item 19)...........lb.
(38) Water evaporated per pound of dry coal (Item 26÷Item 21)........lb.
(39) Equivalent evaporation from and at 212° per pound of coal as fired (Item 28÷Item 19)...lb.
(40) Equivalent evaporation from and at 212° per pound of dry coal (Item 28÷Item 21)...lb.
(41) Equivalent evaporation from and at 212° per pound of combustible (Item 28÷Item 23)...lb.

Efficiency

(42) Calorific value of 1 pound of dry coal by calorimeter............B.t.u.
(a) Calorific value of 1 pound of dry coal by analysis.........B.t.u.
(43) Calorific value of 1 pound of combustible by calorimeter.........B.t.u.
(a) Calorific value of 1 pound of combustible by analysis......B.t.u.
(44) Efficiency of boiler, furnace and grate

$$\left(100\times\frac{\text{Item } 40\times 970.4}{\text{Item } 42}\right)$$ ....per cent

(45) Efficiency based on combustible

$$\left(100\times\frac{\text{Item } 41\times 970.4}{\text{Item } 43}\right)$$ ....per cent

Cost of Evaporation

(46) Cost of coal per ton of ......pounds delivered in boiler room.........
................dollars
(47) Cost of coal required for evaporating 1000 pounds of water under observed conditions.......................................dollars
(48) Cost of coal required for evaporating 1000 pounds of water from and at 212°...dollars

Smoke Data

(49) Percentage of smoke as observed...........................per cent
(a) Weight of soot per hour obtained from smoke meter.....per cent

Firing Data

(50) Kind of firing, whether spreading, alternate or coking...............
(a) Average thickness of fire...................................in.
(b) Average intervals between firings for each furnace during time when fires are in normal condition.......................min.
(c) Average interval between times of leveling or breaking up.......
............min.

(51) Analysis of dry gases by volume
- (*a*) Carbon dioxide (CO_2)....................................per cent
- (*b*) Oxygen (O)....................................per cent
- (*c*) Carbon monoxide (CO)....................................per cent
- (*d*) Hydrogen and hydrocarbons....................................per cent
- (*e*) Nitrogen, by difference (N)....................................per cent

(52) Proximate analysis of coal

	As fired	Dry coal	Combustible
(*a*) Moisture.......	....	....	....
(*b*) Volatile matter..	....	....	....
(*c*) Fixed carbon....	....	....	....
(*d*) Ash............	....	....	....
	100 per cent	100 per cent	100 per cent

(*e*) Sulphur, separately determined referred to dry coal......per cent

(53) Ultimate analysis of dry coal
- (*a*) Carbon (C)....................................per cent
- (*b*) Hydrogen (H)....................................per cent
- (*c*) Oxygen (O)....................................per cent
- (*d*) Nitrogen (N)....................................per cent
- (*e*) Sulphur (S)....................................per cent
- (*f*) Ash....................................per cent

100 per cent

(54) Analysis of ash and refuse, etc.
- (*a*) Volatile matter....................................per cent
- (*b*) Carbon....................................per cent
- (*c*) Earthy matter....................................per cent

100 per cent

- (*d*) Sulphur, separately determined..........................per cent
- (*d*) Fusing temperature of ash..................................deg.

(55) Heat balance based on dry coal.

	Dry Coal	
	B.t.u.	Percent
(*a*) Heat absorbed by the boiler (Item 40×970.4)...		
(*b*) Loss due to evaporation of moisture in coal.....		
(*c*) Loss due to heat carried away by steam formed by the burning of hydrogen................		
(*d*) Loss due to heat carried away in the dry flue gases		
(*e*) Loss due to carbon monoxide.................		
(*f*) Loss due to combustible in ash and refuse.......		
(*g*) Loss due to heating moisture in air............		
(*h*) Loss due to unconsumed hydrogen and hydrocarbons, to radiation and unaccounted for....		
(*i*) Total calorific value of 1 pound of dry coal (Item 42)............................		100

CHAPTER V

BOILER ACCESSORIES AND AUXILIARIES

41. Grates. — Grates are used to support the fuel in a furnace. Most grates are made of cast iron, which is cheap and less liable than other convenient materials to be distorted or twisted under the high temperatures to which it is subjected. The grate must be strong enough to support the load placed upon it, and it must be of such a form that sections can easily be replaced when broken or burned out. It must have sufficient opening for the admission of air to the fuel. The openings or air spaces depend upon the kind of fuel used. The combined area of the openings will usually be from 30 to 50 percent of the total area.

The area of the grate depends upon the amount of coal to be burned and the rate of combustion. Under natural or chimney draft, from 10 to 25 pounds of coal can be burned per square foot of grate surface per hour. Under forced draft, from 40 to 130 pounds of coal may be burned per hour. If hand firing is employed, the grate must not be longer than the distance the fireman can throw the coal accurately (six or seven feet). Depending upon the fuel, draft, and economy of the boiler, the equivalent evaporation per pound of coal will vary from 5 to 12.

Various forms of grates are used. For hand firing, plain grates and shaking or dumping grates are used. The plain grate is harder to keep clean than a dumping grate. Moreover, it is necessary to keep the fire-doors open while the cleaning is in process. The grates used in mechanical stokers are of various types; some are stationary, others traveling and rocking. Occasionally grates are water-cooled, to prevent their burning out. Since this water is led to the boiler after becoming heated in the grate, the boiler capacity is increased, but in most cases the extra care and cost are prohibitive.

42. The Plain Grate. — The grate bars shown in Figs. 16 and 16a are of the stationary type. These grates are cast in small sections so that a section may be easily and quickly replaced when it is burned out. The size of the openings in the bars is governed by the size and kind of coal that is to be burned. If

anthracite coal is used, the openings are small. If the coal is bituminous, and if it cakes, the openings should be made large.

43. The Rocking Grate. — A form of rocking grate is shown in Fig. 17. The bars are supported on pivots, and are dumped or rocked by means of a lever from the front of the furnace. Only the largest clinker need be removed from the top, since the rocking action of the bars breaks up most of the clinker that is formed. In case a strong draft is used, as in the locomotive, this type of grate is usually used in order to keep a clean fire, such as is required with a high rate of combustion.

44. Mechanical Stokers. — The first cost of a mechanical stoker is greater than the equipment for hand firing, but it

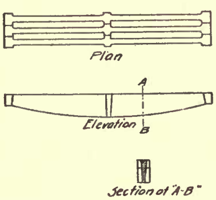


FIG. 16

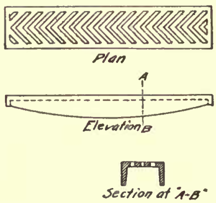


FIG. 16a

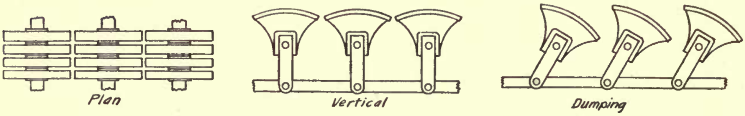


FIG. 17

requires less labor and attendance in its operation. A cheaper grade of fuel can be used, a higher efficiency attained, and less smoke is formed than is usual with hand firing. In a fair-sized or large plant, it is usually better economy to use some form of mechanical stoker. There are many forms of stokers in use.

45. The Chain Grate Stoker. — Where a low-grade fuel is used, as is often the case in the middle west and in the central states, the chain grate (Fig. 18) is extensively used. The grate is composed of a large number of short links, forming an endless chain. This chain runs over front and rear sprockets. Power is used to drive one of these sprockets, causing the whole chain to revolve slowly at a speech which is regulated by a suitable mechanism. The whole grate is mounted on wheels so that it can be run out in the open for repairing and cleaning.

The coal is fed to the front of the grate from a hopper which extends across the entire width of the grate. At the rear of the hopper there is a plate lined with firebrick that may be raised or lowered, thus regulating the depth of fuel-bed. The volatile matter in the fuel is distilled off as the coal first enters the furnace. These volatile products pass back over the part of the fire where the fixed carbon is burning, and are given a chance to burn there. By the time the fuel-bed has reached the rear of the furnace the combustion should be complete. The ash and clinker are dropped off to the ash-pit at the rear.

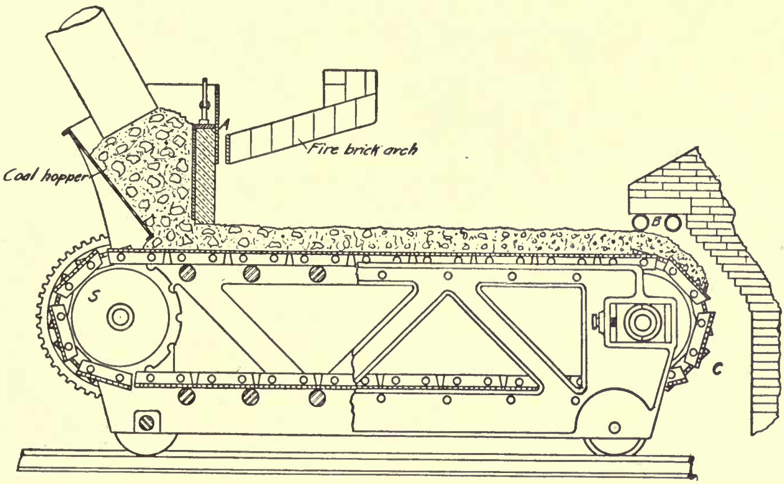


FIG. 18

The front part of the grate is overhung by a firebrick arch. This allows sufficient time and temperature for complete combustion before the gases strike the heating surface of the boiler.

Incomplete combustion is apt to occur with a chain grate if the fire is forced very hard. Excess air is likely to leak in if the fuel-bed becomes too thin. This causes a drop in the temperature of the combustion chamber and therefore poor combustion.

Under a light load, the fuel is often burned before it reaches the rear of the grate. Air is likely to leak through the ash, causing poor economy. As a remedy for this condition, a contrivance similar to a damper is sometimes placed under the rear portion of the grate. This makes it possible to shut off the air supply from this part of the grate.

46. The Roney Stoker. — The inclined grate stoker is one in which the coal is fed from a hopper at the top, the coal burning

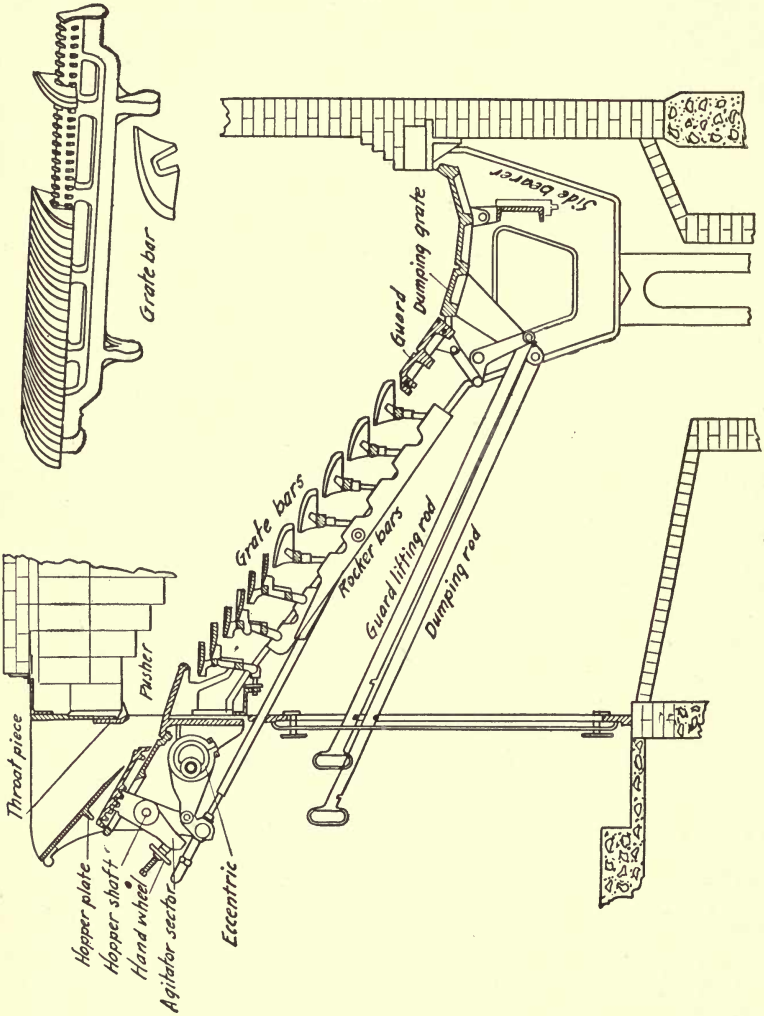


Fig. 19

on its way down across the sloping grate. In the forms customarily used, the grates are operated mechanically. There are two classes of these grates, side-feed and front-feed.

Figure 19 shows the Roney type of front-feed stoker. The coal is fed into the hopper, usually by gravity from bins above. A reciprocating pusher forces the coal from the hopper onto a dead plate beneath the front of the arch, where the distillation starts. From this plate, it is made to move downward by the motion of the grates. By the time the fuel reaches the ash plate at the bottom of the incline the combustion is complete. The ash is dumped into the pit below from time to time by means of a hand-lever that is operated from the front of the furnace. The grates are rocked by means of an eccentric placed on a rotating shaft running horizontally along the front of the whole battery of boilers. The amount the grates are rocked, and the amount of coal fed, are under the control of the fireman.

Since the distilled products are driven off at the front of the firebrick arch, they have time, and are at such a temperature, due to the fire below, that complete combustion takes place. In some cases, steam and air are admitted under the front of the arch to aid in the combustion. The length of arch varies with the grade of fuel to be used, and with the kind of boiler under which the stoker is installed. In some cases it covers the entire grate, forming a Dutch oven which sits out in front of the rest of the boiler setting.

47. The Underfed Furnace. — In the types of stokers previously described, the volatile matter is distilled and burnt over the bed of burning fixed carbon. As the feeding of fuel is uniform the amount of gas given off at any one time is not so great as in hand firing. Hence combustion has a chance at all times to be more nearly complete. Another and radically different method is to feed the green coal from beneath, blowing the volatilized matter along with sufficient air up through the hot fuel-bed above, where complete combustion takes place.

Figure 20 shows such a stoker. The coal is fed into a hopper and is forced back under the fuel-bed into *retorts*, by means of a ram or plunger. Air under pressure is forced in through tuyeres at the distillation zone, and by the time the gases pass through the hot fire of fixed carbon and reach the top of the fire, they are completely burned. This type requires a forced draft. The refuse is forced back and down onto the dump plates at the rear of the wind-box, and is dropped from there into the ash-pit below through an adjustable opening.

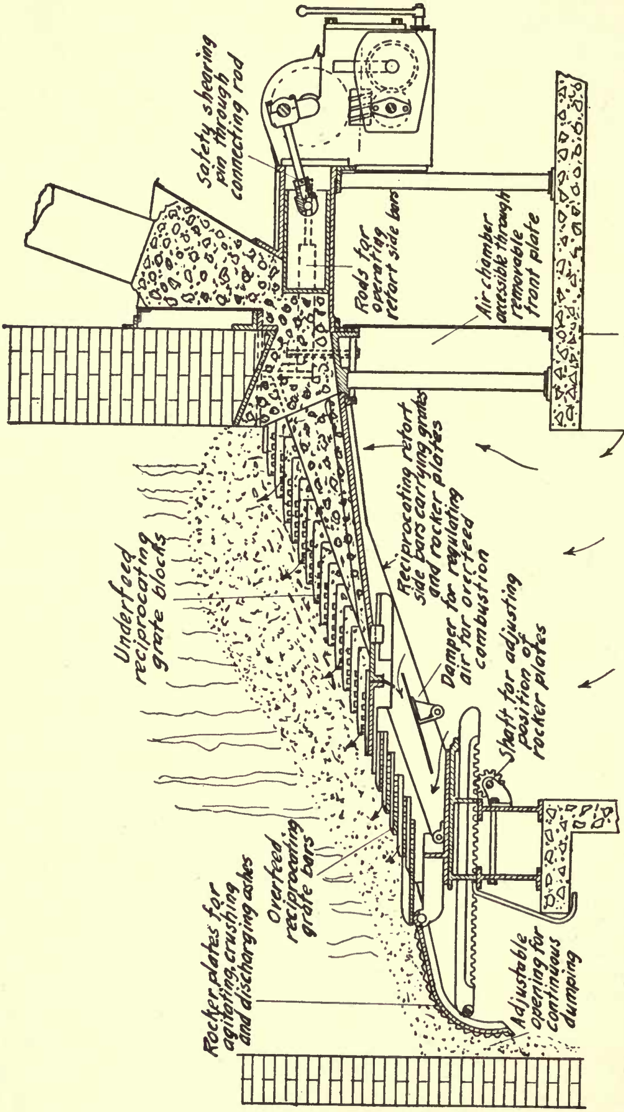

Safety shearing pin through connecting rod
Rods for operating retort side bars
Air chamber accessible through removable front plate
Reciprocating retort side bars carrying grates and rocker plates
Damper for regulating air for overfeed combustion
Shaft for adjusting position of rocker plates
Underfeed reciprocating grate blocks
Overfeed reciprocating grate bars
Rocker plates for agitating, crushing and discharging ashes
Adjustable opening for continuous dumping

FIG. 20

Since the temperature of the fire is very high, most of the ash is fused. In order to prevent clogging it is sometimes necessary to have a water-cooled bridge wall.

Due to the forced draft, a plant thus equipped is not subject to the variations due to changing weather conditions. The rate of combustion is regulated by the amount of coal fed and the amount of air blown in, which are controlled together. The underfeed type of stoker therefore is more flexible and will give higher efficiency under conditions of forced load than those previously described.

48. Smoke Prevention. — Soft coal is generally considered to be smoky. Nevertheless, it is possible to burn practically all grades of bituminous coal with very little smoke. Only during the past few years has there been any very determined effort to rid ourselves of the smoke nuisance. Several of our larger cities have ordinances which are enforced more or less rigidly against the excessive emission of black smoke by power plants.

The soot, which is the solid and black part of the smoke, disfigures buildings and may even injure health by keeping out the sunlight and by clogging up the respiratory organs. It is frequently stated that a great amount of fuel goes to waste in the soot of the smoke. While the soot may be a nuisance, yet it represents in itself but little heat loss. Soot indicates, however, incomplete combustion, which often means that there is a large amount of unburned hydrocarbon and probably some CO that should be burned to CO_2. Smoke prevention, in many cases, results in an increased economy of the power plant. Thus we often find instances of plants that have installed modern equipment which prevents the formation of smoke, not so much with the idea of eliminating the smoke as to obtain better efficiency by means of proper combustion.

Let us consider in a general way the causes of smoke production. Bituminous coal, upon being heated to moderate temperatures (600° to 1000° F.), will have certain hydrocarbons driven from it in a volatile state. This volatile matter will burn when mixed with a sufficient amount of air and raised to a sufficiently high temperature (1800° to 2000° F.), forming carbon dioxide and water. If for any reason there is an insufficient supply of oxygen, or if the hydrocarbons are not raised to a sufficiently high temperature, incomplete combustion will follow, and black smoke

may result. The fixed carbon left in the coal after the volatile hydrocarbons have been driven off, with the addition of sufficient air, will burn to carbon dioxide. If there is a lack of air, carbon monoxide, or a mixture of carbon monoxide and carbon dioxide, will result.

In conclusion, to burn bituminous coal smokelessly, a furnace must have a sufficient supply of air to insure the complete combustion of the volatile matter, and it must have a temperature high enough to permit of this combustion. In the ordinary furnace the time taken for the gases to pass from the grate to the comparatively cool heating surface of the boiler, where they are rapidly cooled, is quite small, perhaps less than a second. At least, not enough time is allowed for the volatile matter to unite properly with the oxygen of the air, and black smoke is the result. If the path of the hot gases can be made longer, thus giving them time to burn, a large reduction can be made in the amount of smoke. It is sometimes possible to rearrange the baffling in the path of the gases in such a way that this is accomplished easily. Another way to lengthen the time of burning is to move the grates out from under the boiler and place a long firebrick arch over the fire.

In ordinary hand firing, a large quantity of cold coal is thrown on top of the fire, with the result that the fire is greatly cooled, both to heat up the coal and also to cause distillation of the volatile matter. This reduces the temperature to a point at which the complete combustion of the volatile matter will not take place, and smoke results. At the time the volatile matter is given off an excess of air is needed to burn it. If this air is let in over the top of the fire it will still further cool the fire, which only adds to the trouble.

The use of the various over-feed stokers, such as the chain grate, and those with the front or side feed, is a decided improvement over hand firing because the fresh coal is added continuously, and the air supply can be properly adjusted and maintained. In these types of stokers, the fresh coal, upon coming to the furnace, passes through the distillation period in such a position that the volatile products must pass over the fire of fixed carbon and be burnt there.

In the underfeed type of stoker, the fresh coal is forced in from the under side of the fire, and the distilled products along with sufficient air to burn them, are forced to pass through the hot

bed of burning carbon above, with the result that a sufficiently high temperature is maintained to allow for their complete combustion.

The purpose of the down-draft furnace is much the same as that of the underfeed stoker. In this type, the zones of distillation and of burning the fixed carbon are separated. The first takes place on the upper grate and the second on the lower; the distilled product passes over the hot fuel bed on the lower grate and complete combustion occurs.

Many devices to prevent smoke are on the market. Most of them consist of some form of steam jet that carries in and mixes a sufficient amount of air with the volatilized hydrocarbons to effect complete combustion. The steam itself has no power to prevent smoke. It is used simply to carry the air and to mix it with the volatile products. If the steam jet is left on too long after the period of distillation, a loss greater than the gain effected by the jets may result. Some makers use a dash-pot or other arrangement that automatically shuts the steam off soon after each firing.

49. Settings. — The brickwork that surrounds a boiler is called a ***setting.*** The outer side of this setting is built of common red brick. The inner surfaces that are exposed to the high temperature of the flame are lined with firebrick. With the very high temperatures that exist in modern furnaces, it is difficult to get a grade of firebrick that will give satisfactory service. It is better practice not to leave an air space between the outer wall and the lining, since heat is transmitted through the air space, under the high temperatures that exist in a furnace, faster than it would be through the same thickness of brick.

In most furnaces, a firebrick arch is placed over the fire. This arch forms a chamber in which the temperature is kept very high. The length of this arch varies with the kind of fuel to be used and with the type of boiler. Firebrick baffles are placed between the tubes of water tube boilers in such a manner that the gases are forced to pass the heating surface several times on their way to the stack. The burnt gases pass from the setting to the stack through a duct called the breeching.

Since there is a difference in pressure between the outside and inside of the setting, it is important that there be no cracks for the air to leak in or for the gases to leak out. Under natural

draft, there is a leakage of air inward, which cools the boiler and injures the draft. With a forced draft, the gases may leak out into the boiler room.

50. Draft. — Natural draft is obtained by means of a stack or chimney. The gases as they leave the boiler are at a temperature of from 400° to 600° F. At this temperature they are much lighter than the outside air. Since the column of gas in the stack is lighter, it is forced up from the bottom by the heavier air outside. The stack should be large enough so that but little draft will be lost by the friction between the gas and the stack. It should be insulated so that little heat is lost through the walls of the stack.

There are three kinds of stacks in use, steel, brick, and concrete. The steel stack is cheaper and lighter, but it is expensive in its upkeep, since it must be painted often to prevent the corroding of the plates. In the better grades of steel stacks, a firebrick lining is used at least a part of the way up to prevent conduction of heat to the outside. Brick stacks are sometimes built of hard common brick, but of late years more often of special radial brick. They are sometimes lined with firebrick, as in the steel stack. Reinforced concrete stacks have come into use during the past few years. When properly put up, they give good service.

Brick and concrete stacks must be heavy enough to resist the overturning effort of the wind. Steel stacks are either anchored and designed to withstand the bending action due to the wind, or else they are supported by guys.

Where forced draft is used, the stack need be only high enough to discharge its smoke above the surrounding buildings. Forced draft is obtained by means of fans or blowers which force the air into the ash-pit or wind-box and thence through the fire. In locomotives, the forced draft is obtained by means of nozzles through which the exhaust steam from the engines is discharged into the chimney. A draft caused by this method is sometimes called induced draft.

The amount of draft is measured in inches of water. Under natural draft it will increase in going from the ash-pit to the stack and at the base of the stack it will be from 0.5 to 1.5 inches. Under forced draft the pressure in the ash-pit is greatest, and will vary from 1 to 5 inches.

51. Dampers. — A damper should be interposed between each furnace and the stack. The efficient operation of the furnace necessitates careful attention to the damper. Automatic damper regulators are in use, but for the best results they should be supplemented by intelligent manual control.

52. Safety Devices. — There are in general three causes of explosions of properly designed boilers: a weakened part, high pressure, and low water. The first is due to the corroding or wearing away of some part of the structure, or to a local overheating due to an accumulation of sediment or scale. It may be due to an undetected flaw in the materials entering into the makeup of the boiler, or it may be due to carelessness or poor workmanship during construction.

The second cause, high pressure, is due to a pressure much in excess of that for which the boiler was designed. This may be due to a faulty safety valve, or to the ignorance of a fireman.

The third cause, low water, allows some of the parts to get overheated and therefore much weakened. It may exist unknown to the fireman on account of foaming, which is liable to cause an untrue indication of the water level in the gage glass, or on account of some stoppage in the connection to the glass.

To safeguard against accidents due to a weak part, it is necessary to have a thorough inspection both of materials entering into construction of the boiler and of the workmanship during construction. There also should be frequent inspection after the boiler is put into service. The common test for strength is hydrostatic. Before being put into service a boiler should have water pumped into it, and a pressure should be reached much in excess of the working pressure.

53. The Pressure Gage. — The pressure that exists in a boiler is measured by a steam gage. In this country, the dial of the steam gage is graduated to read in pounds per square inch. The gage almost universally used is known as the Bourdon gage. Figure 21 shows its internal construction. The pressure is admitted to a curved flattened tube which is closed at its free end. This internal pressure tends to make the curved tube straighten out. The free end is connected to the needle by means of levers, a rack, and a pinion. Any movement of the free end causes the hand or needle to turn, and the pressure causing the movement is indicated on the properly graduated dial. The flattened tube

is made of brass or steel. Since a change in the temperature of the tube would cause an error in the reading of the gage, it should be connected to the boiler or steam-pipe by means of a siphon so that steam may never enter the gage. On locomotives, where the gage is subject to continual and severe jarring, two stiffer tubes are used in place of the one shown in the figure. The better grades of gages have a light hairspring to take up the backlash in the levers and gears.

A gage similar to the one shown in Fig. 21 is often used to indicate vacuum. For a vacuum, the tube is bent still more and the levers are so arranged that the motion of the needle is reversed from that of the one shown in Fig. 21. The dials of vacuum gages are commonly graduated to read in inches of mercury. Where pressures are had that may fluctuate from a vacuum to a positive pressure, gages are used that will indicate either the vacuum or the pressure above the atmosphere. Gages should be tested from time to time to see that they give the correct pressure reading.

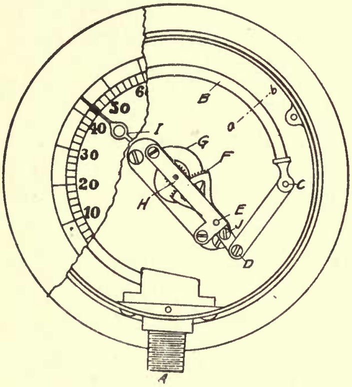


Fig. 21

54. The Safety Valve. — The purpose of a safety valve on a boiler is to prevent an undue or dangerous pressure. It is impossible in an ordinary furnace to regulate the combustion quickly enough to correspond to sudden changes of the amount of steam used. For instance, a boiler may be furnishing its maximum amount of steam, when for some reason the engine is shut down without warning, and therefore before the fire can be deadened. As a result, the rapid formation of steam will continue long enough to cause an excessive boiler pressure if the safety valve does not give relief. Hence the safety valve should have such a capacity that it is capable of discharging all the steam that the boiler can generate without allowing the pressure to become dangerous. Furthermore the safety valve must be absolutely reliable in its action, and it should be so constructed and placed on the boiler that it cannot be put out of action through carelessness or ignorance. No stop valve should be placed between it

and the boiler. Many explosions have been caused by the failure of the safety valve to operate. As far as the writer knows, however, none have occurred that were due entirely to excessive pressure when the valve was in action. Several types of safety valve have been used in the past, but with the pressure ordinarily carried in this country, the use of the *pop* type has become almost universal, and will be the only one described here.

Figure 22 shows in section a ***pop safety valve.*** Most safety valves are made with a 45° seat. The valve is held on the seat by a helical spring. When the steam pressure becomes sufficient to overcome the force of the spring, the valve is raised enough to allow some steam to escape. This steam passes into a *huddling chamber.* The area upon which the steam now acts is slightly greater than before the valve opened, with the result that the spring is compressed suddenly to a greater extent than if the steam acted only upon the original area. Escape of steam will continue until the pressure has dropped to a few pounds less than that at which it opened. When the valve stops blowing, it seats firmly. Since the pressure is less than that at which it opened, it will remain shut until the steam pressure again reaches the popping point. The valve should be constructed and set so that the difference between the popping pressure and the closing pressure or *blowdown* is not too large, in order to prevent shock to the boiler and an excessive loss of steam during ordinary operation.

A lever is provided at the top of the casing for locking the valve open. The compression in the spring may be adjusted by screwing the top cap up or down. In most valves the spring is encased so as to protect it from the escaping steam, and to prevent back pressure in the discharge pipe from acting on the top of the valve. (See Fig. 22.)

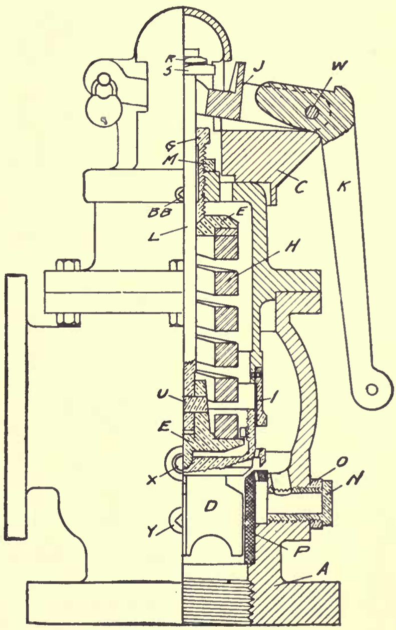


FIG. 22

55. Safety-valve Capacity. — The amount of steam that a safety valve discharges depends upon the steam pressure and upon the effective opening. The latter varies with the lift of the valve. Not all valves of the same diameter have the same lift; hence they differ in capacity. There has been considerable agitation recently to have all pop valves put upon a uniform *rating*. Tests have been made to determine the discharge and the lift under various conditions of pressure. These tests show that the commonly used empirical formula given by Napier is substantially correct when applied to the safety valve. $W = \frac{AP^{.97}}{60}$

56. Napier's Formula. — Napier's formula for steam issuing from an orifice into the atmosphere is

$$W = \frac{A \cdot P}{70}$$

in which W is the weight in pounds of steam issuing per second, A is the area of the orifice in square inches, and P is the absolute pressure in pounds per square inch in front of the orifice.

Applying this formula to the safety valve with a 45° seat, it is seen that the area of opening, A, equals approximately the product of πD and the lift times the sine of 45°, where D is the diameter of the valve in inches. If the lift is known, the discharge may be calculated. Some valve makers use the assumption that the lift is 1/30 of the diameter, and rate their valves accordingly. Under this assumption it is seen that

$$W = \frac{\pi D^2 \times .707 \times P}{70 \times 30},$$

and the weight of steam discharged per hour is $3600\ W = 3.81\ PD^2$.

As previously explained, the safety valve must be large enough to discharge the maximum amount of steam the boiler is capable of generating. We can compute this maximum amount from the heating surface of the boiler, allowing an evaporation of from six to ten pounds of water per square foot of heating surface per hour, or we may compute it from the grate area, assuming a boiler efficiency and a rate of combustion consistent with the draft and with the kind of fuel used. The former method is considered better. After conducting numerous tests P. G. Darling[1]

[1] Trans. A. S. M. E., vol. 31 (1909), p. 109.

advocated to the A. S. M. E. formulas for pop safety valves derived according to the following method:

$$\text{for stationary boilers, } D = 0.068\,\frac{H}{LP},$$

$$\text{for locomotive boilers, } D = 0.055\,\frac{H}{LP},$$

in which D is the diameter of the valve in inches, H is the heating surface of the boiler in square feet, L is the lift of the valve in inches, and P is the absolute boiler pressure in pounds per square inch. It is noticed that smaller valves are required for locomotives, because the maximum draft can be secured only when the steam is being drawn from the boiler by the engine.

57. Other Safety-valve Formulas. — Various cities and states have their own rules governing the sizes of safety valves, a few of which are given below.

City of Chicago. One square inch of pop-valve area ($\pi D^2/4$) for every three square feet of boiler grate area.

City of Philadelphia. For pop valves, $A = 22.5 \times G/(p - 8.62)$, in which A is the area of the valve in square inches (not the effective opening for the escape of steam), G, the grate area in square feet, and p the gage boiler pressure.

U. S. Supervising Inspectors. $A = .2074 \times WH/P$, in which A is the area of the valve as in the previous formula, WH is the number of pounds of water evaporated by the boiler per hour, and P is the absolute boiler pressure.

Safety valves are not made in sizes over 5 or 6 inches in diameter. In large boilers it is therefore necessary to use more than one.

Example. What should be the size of pop safety valve on a boiler with 1500 square feet heating surface if the pressure carried is 130 pounds gage?

Solution. Assuming a maximum rate of evaporation of 8 pounds of water per square foot of heating surface per hour, we get $8 \times 1500 = 12000$ pounds of steam to be discharged per hour through the valve. The weight discharged per second is $12000/3600 = 3.33$ pounds. From Napier's formula, $W = AP/70$, we see that $3.33 = A \times (130 + 15)/70$, whence A, the area of opening, in square inches, is 1.61 Assuming a lift of valve equal to 1/30 of the diameter, the area of the opening will be approximately $.707\pi D^2/30$. Then $.707\pi D^2/30 = 1.61$, or $D = 4.67$ inches. Hence a 5-inch valve should be used.

58. The Water Glass or Gage Glass. — In order that the amount of water in the boiler may be known, a water glass is attached. The lower end of the water glass is attached to the

water space, and the upper to the steam space. Since there is danger of the glass becoming stopped in these connections, and the water level thereby being falsely indicated, or of the glass being broken, three gage-cocks or try-cocks are placed on the boiler or water column. The top cock is placed above, and the bottom one below, the normal water level. By opening these cocks in succession, one may determine whether or not the gage glass is giving the correct level.

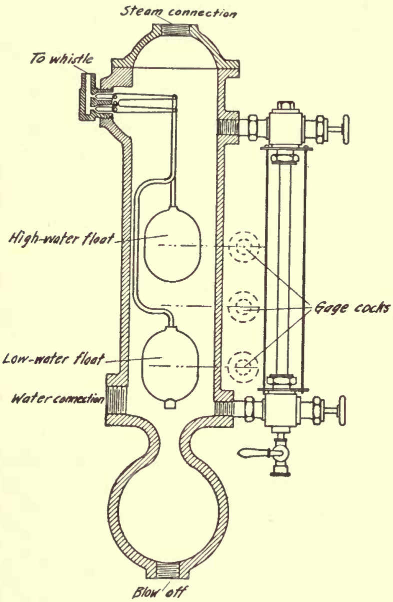


FIG. 23

59. High and Low Water Alarm. — High-water and low-water alarms are sometimes used to attract the attention of the fireman when the water falls below or rises above the safe level. The alarm is operated by means of a float in the water column. When this float rises too high or falls too low it will open a valve, and allow the escaping steam to blow a whistle. (See Fig. 23.)

60. Fusible Plug. — Another safety device used to detect low water is the fusible plug. This plug (Fig. 24) has a tin core that will melt when the water level falls below it. These plugs are placed in the crown sheet of an internally fired boiler in the rear head a little above the top tubes in the return-tubular boiler and in the bottom of the steam drum of a water-tube boiler. They should be kept free from scale on the inside and from soot on the outside. None of the above safety devices are absolutely certain in their action. Under conditions of very rapid steaming or with feedwater that foams, the water level in the boiler may be below that in the water column.

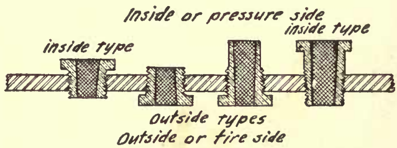


FIG. 24

61. Boiler Feedwater Treatment. — The impurities in water that are responsible for most of the scale formation are the carbonates and sulphates of calcium and magnesium. If muddy water is used, the mud may be deposited on the heating surface and aid in scale formation.

The carbonates of lime and magnesium are soluble in water containing carbon dioxide. These carbonates cause what is known as *temporary* hardness.. Upon heating to 212° F. the carbon dioxide is driven off, and the carbonates are precipitated. If these are the only impurities in the feedwater, an open feedwater heater will remove most of the scale-forming material. Where a heater is not used the carbonates may be precipitated by the addition of a solution of slacked lime. The lime combines with the carbon dioxide to form the insoluble monocarbonate of lime.

The sulphates of lime and magnesium are not precipitated at a temperature of 212°, but are precipitated at a temperature such as exists in the boiler. They cause what is known as *permanent* hardness. The addition of carbonate or hydrate of soda (or a mixture of the two) will cause precipitation. The carbonate of soda decomposes the sulphates and forms insoluble carbonates of lime and magnesium, which precipitate, leaving neutral soda and sodium sulphate in solution. If carbon dioxide is present, the soluble bicarbonate of lime is formed, which may be precipitated by heating or by the addition of lime as explained previously. In most purification processes both the lime and soda are used.

If organic matter, from sewage or from some other source, is present in the water, it may be removed by filtration. Before passing the filter a coagulant, such as alum, is often used. Organic matter in feedwater is often the cause of foaming.

62. Scale Prevention and Removal. — Many substances have been used to prevent the formation of scale. Some of these probably do as much damage to the boiler as would the scale. Aside from the treatment to remove the scale-forming material, the best substance seems to be graphite. When it is injected into the boiler, it is said to help in the prevention of scale formation.

Where no precaution is taken to prevent the scale from forming, it is necessary to clean it from the tubes periodically. This is usually done by means of a cutter or hammer that is driven by a small air, steam, or water turbine.

Figure 25 shows one make of cleaner that is applied to a fire tube. Figure 26 shows the form that is applied to a water tube.

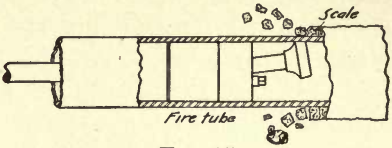


FIG. 25

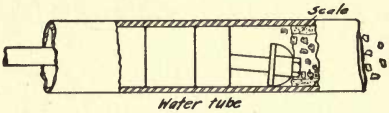


FIG. 26

63. Oil Separators. — In plants where all or part of the steam is condensed and used again as boiler feed, the oil that was used to lubricate the engine will find its way to the boiler. This does not apply to steam turbines, as oil is not usually used internally with them. This oil forms a very hard scale that it is almost impossible to remove. To prevent this, oil separators are used to remove the oil, either from the exhaust steam or from the water after it is condensed. The removal before condensation is preferable, since the oil does not have to be contended with in the condenser or feedwater heater. Figure 27 shows an Austin oil separator. Since the separator is quite large, the steam passes through it with a small velocity and deposits the oil on the surface of the corrugated vertical baffle plate shown in plan and section in the figure. With high vacua, a spray of water keeps the surface moist, which aids in the separation of the oil.

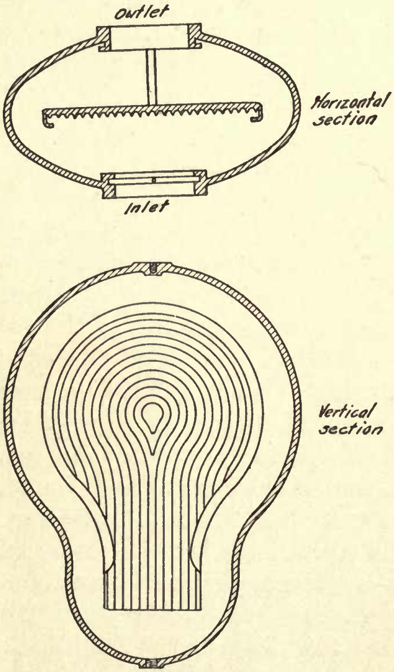


FIG. 27

64. Boiler Feed-pumps. — The feedwater is usually forced into a boiler by means of a pump. Figure 28 shows a common type of boiler feed-pump. This pump is direct acting, the steam piston and water piston or plunger being fastened to the same piston rod. Since the steam and the water in a boiler are both under the same pressure and since pipes and fittings offer a resistance to the flow of both water and steam, it is seen that it is necessary to

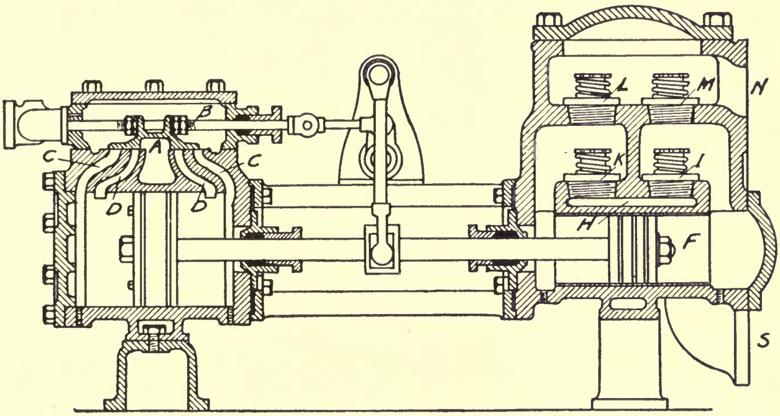


FIG. 28

make the steam piston of the feed-pump considerably larger in diameter than the water plunger.

Steam is admitted by the steam valve alternately to the two ends of the steam cylinder. At the same time that the valve is admitting steam to one end of the cylinder it is allowing it to exhaust from the other, thus giving the piston a reciprocating motion. The water piston sucks up water from the suction pipe in one end of the water cylinder while forcing it into the delivery pipe on the other.

The suction pipe leads either to the hot well or to the cold well from which the feedwater is taken. The lower end of the suction pipe should be provided with a strainer to prevent any large pieces of solid matter from getting into and clogging the valves. A foot valve should be provided also to keep the suction pipe

full of water when the pump is not running, thus eliminating the priming of the pump every time it is started.

There are two sets of valves at each end of the water cylinder. On the suction stroke, the suction valves are lifted and the water is sucked in back of the piston. During the forcing stroke, these valves are closed and the water is forced out through the upper set and into the delivery pipe. A light spring is employed to help seat the valve. Most valves are faced with a composition disc which may be replaced when it becomes worn.

The flow from a reciprocating pump is not steady; to insure a more uniform rate of flow of the water an air chamber is placed on the delivery line close to the pump. This is kept partly filled with air, which acts as a cushion. A check valve is placed in the feedline between the pump and the boiler. This prevents the water from the boiler escaping back through leaky valves when the pump is not in full operation. There should also be a stop valve in the feedline.

There are two types of reciprocating steam pump, one in which there is only a single steam cylinder and a single water cylinder, and the other in which there are two steam cylinders and two water cylinders placed side by side. The latter type is called a duplex pump. In this type the valve for one steam cylinder is operated by the movement of the piston of the other steam cylinder.

These boiler feed-pumps take steam the full length of the stroke, not allowing it to expand in the cylinder, and they are not economical in the use of steam. (See Chapter VI on the steam engine.) However, only a small proportion of the steam generated by the boiler is needed to run the pump. To secure better economy, feed-pumps are occasionally driven by power taken from the main engine. The supply of water from these pumps is not easily regulated. They are made to pump more water than is normally required, the excess being passed back to the suction through a relief valve. In large electric power plants, triplex pumps driven by electric motors are often used for boiler feeding. Of late years centrifugal and turbine pumps have been employed for boiler feeding.

An automatic regulator is sometimes installed with the pump so that the pump will furnish just the proper amount of water to keep the boiler water-level constant.

65. The Injector. — On portable boilers and in small plants, the water is often forced into the boiler by means of an injector or inspirator. This is usual also on locomotives. The principle upon which the injector works is illustrated by Fig. 29. Steam from the boiler enters the injector through a steam nozzle, *a*, in which it expands and some of its heat energy is transformed into kinetic energy. The steam leaves the nozzle with a high velocity and enters a small combining tube, *b*. The water inlet leads to a chamber which is located between the nozzle and the combining tube. As the steam flows from the nozzle to the combining tube it tends to form a partial vacuum in the water chamber and thus sucks up and carries the water along with it. The steam mixes with the water in the combining tube and is condensed. This mixture of condensed steam and water has a high velocity and therefore a considerable amount of kinetic energy; its pressure, however, is atmospheric or less. This mixture passes from the combining tube to the delivery tube, *c*, which has an increasing diameter. The mixture therefore loses a large amount of its velocity and kinetic energy. What it loses in kinetic energy it gains in pressure energy, so that by the time it leaves the injector it has gained enough pressure to force open the check valve leading to the boiler. An overflow is located at the end of the combining tube, so that when steam is first turned on, it escapes through the overflow. The overflow is fitted with a valve which automatically closes when the pressure inside the combining tube falls below the pressure of the atmosphere, thus preventing air from coming into the injector and impeding its action.

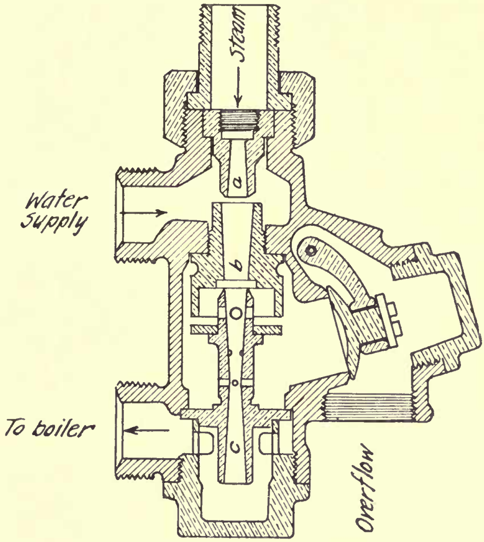


Fig. 29

Unless specially constructed, an injector cannot lift water a very great height. Moreover, since the injector must condense the steam in order to work at all, it is necessary that the water be cold. Considered as a pump, the efficiency of the injector is very low, because the greater part of the energy of the steam goes to heat the water. If it is used to feed a boiler, the heat spent in raising the temperature of the feedwater is not lost, as it goes back into the boiler. Hence it is efficient for this purpose. The injector is light, occupies but little space, and is cheaper than a pump, but it is not so dependable.

66. Boiler Feeding by Return Trap. — The condensation from various parts of the plant is sometimes returned to the boiler by what is known as a return trap. This trap is located above the level of the boiler and the water runs into the boiler under the influence of gravity and the pressure of live steam. These traps are quite economical in the use of steam and they may be used to supply all the feedwater. They are not nearly so reliable as the pump or injector, however, and are therefore but little used to furnish the entire feedwater supply.

67. The Steam Line. — Steam pipe is made of wrought iron or of steel. The nominal diameter corresponds approximately with the inside diameter. Sizes of standard pipe vary, by the 1/8″ from 1/8″ to 1/2″, by the 1/4″ from 1/2″ to 1 1/2″, by the 1/2″ from 1 1/2″ to 5″, and by the 1″ from 5″ to 15″.

It has been customary to allow an average velocity of steam in the line of from 4000 to 6000 feet per minute. In modern turbine plants, however, where the flow is uniform, and especially where superheated steam is used, velocities much in excess of these values are used. If a velocity of wet steam much greater than that just mentioned is used, the drop in pressure due to skin friction will be excessive. On the other hand, if a velocity much less is allowed, too large and expensive a pipe will be required.

If the volume and velocity of steam to be carried by the pipe line are known, the diameter is easily determined. The volume carried per unit of time equals the product of the area of the cross-section of the pipe and the velocity. If the size and speed of the engine to be supplied are known, we may compute the volume of steam needed. At maximum it may be assumed that the engine takes steam during the full length of the stroke. When

more than one boiler is used it is customary to discharge the steam into a common pipe called a header. In such a case each boiler should be provided with a *non-return* stop-valve between the boiler and the header. This non-return stop-valve acts as a check valve in case the direction of steam flow should be reversed, which would happen in case a tube blew out or some other similar accident occurred.

The piping must be provided with a sufficient number of hangers to prevent breaking due to its own weight. The line should slope downward in the direction the steam is to flow in order that the condensation may be carried along with the steam. If this precaution is not followed condensed steam will collect in the pipe and may be carried in slugs by the steam in amounts large enough to injure and cause leakage or even breakage of the fittings. Provision should be made at the low points of the line to remove condensation. A pipe should be run down from the low point and the water collecting in this may be blown out from time to time by opening a valve by hand. A trap may be installed that will remove it automatically.

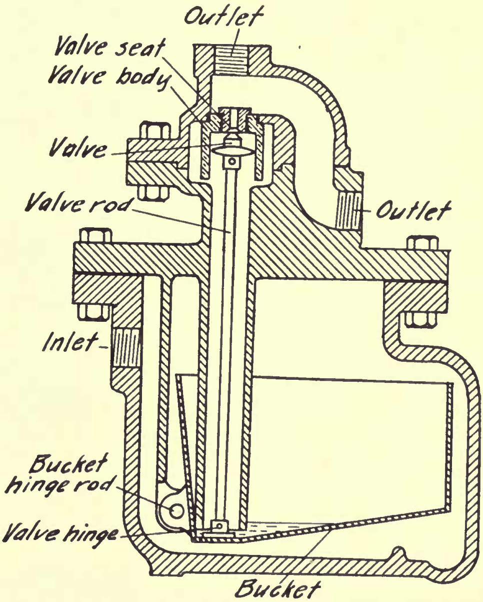


FIG. 30

68. The Steam Trap. — Several types of traps are in use. In the more common kinds, the valve is operated by means of either a float, the unequal expansion of two different metals with changing temperature, pressure of collected water on a flexible diaphragm, or the weight of a bucket as it fills with water. The latter kind is illustrated in Fig. 30. In this type the buoyancy of the bucket keeps the valve closed until enough water flows over the

edge and collects in the bucket to sink it. The sinking of the bucket opens the valve and the water collected in the bucket is forced out through the valve by the steam pressure inside the trap. The bucket now being lightened, it again rises, closing the valve.

In many traps, the valve is operated by a float. The water collects in a float chamber and raises the buoyant float until the valve is opened. The water then escapes until the float is lowered enough to allow the valve to seat. An air valve is located at the top of the trap to allow the air to escape if enough should be caught there to interfere with the operation of the trap.

Another form of trap is one in which the valve is operated by the unequal expansion of two metals. When the trap is cold the valve is open and the water is allowed to escape. As soon as the steam flows through, however, the parts are heated and ex-

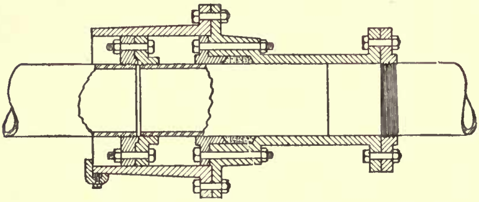


FIG. 31

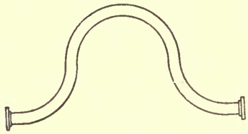


FIG. 32

pand unequally, closing the valve. Water then collects again, and as the parts cool, the valve will again open and the operation will be repeated. When large amounts of water are to be handled, dumping traps may be used. The discharged water from the trap is led to the drain or is piped back to the hot well.

69. Expansion Joints. — Since the pipe is laid cold, it will expand when steam is turned into it and its temperature becomes that of the steam. The expansion amounts to 2.5 inches per hundred feet of pipe with ordinary steam temperatures, and may be greater when the steam is of very high pressure and is superheated. The piping must be so arranged that this expansion may take place without injury to the pipe. If the pipe is not laid straight but contains elbows, it may bend enough so that no dangerous stresses will be induced. If there is a considerable run of straight pipe, however, expansion joints must be provided.

There are several types of expansion joints in use. A very common kind for use with low-pressure steam is the slip-joint. In

this, provision is made for the slippage of one part of the joint on the other. The joint is kept steam tight by means of a stuffing box. Figure 31 shows this type. Goosenecks and expansion loops (Fig. 32) are used when the steam pressure is high.

70. Steam Separators. — Unless superheat is used, steam leaving the boiler will always contain some moisture. If the steam-pipe is very long, some condensation also takes place. Due to these causes, the steam is liable to reach the engine quite wet. It is desirable both for safety and for economy to have the steam as dry as possible when it enters the engine. To remove the moisture from the steam, a separator is placed in the line just before it reaches the engine. The steam is given a sudden change in direction upon entering the separator. The moisture resists this change to a greater extent than does the steam. In the type shown in Fig. 33 the steam is first deflected downward and then upward, and as the moisture cannot change its direction of motion as rapidly as the steam, it is caught and collected in the bottom of the separator.

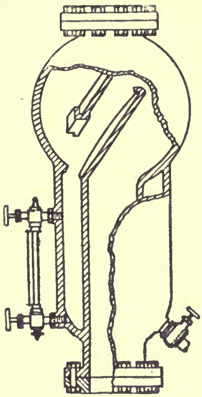


FIG. 33

In some makes the steam is given a whirling motion and the water, being denser than the steam, is forced to the outside of the separator, where it is collected. Another type, which is similar to the oil separator of Fig. 27, is that in which a corrugated baffle plate is interposed in the path of the steam. The steam passes around the baffle while the moisture is caught by it and runs down the corrugations to the bottom of the separator, where it is collected.

A separator should remove most of the moisture, but it should not offer too great a resistance to the passage of the steam, since this would cause a drop in pressure. The moisture, after being collected, is trapped off and discharged to the drain or returned to the hot well. Often the separator is made large and acts as a steam receiver. This reduces the pulsation in the steam line when the steam is used by a reciprocating engine.

71. Steam-pipe Covering. — To prevent radiation of heat from the steam-pipe and the consequent condensation, a covering is applied to the pipe. The covering is made from materials that

are poor conductors of heat. A finely-divided, dead air space is one of the best non-conductors of heat. In most coverings the object is to get as much finely-divided dead air space as possible.

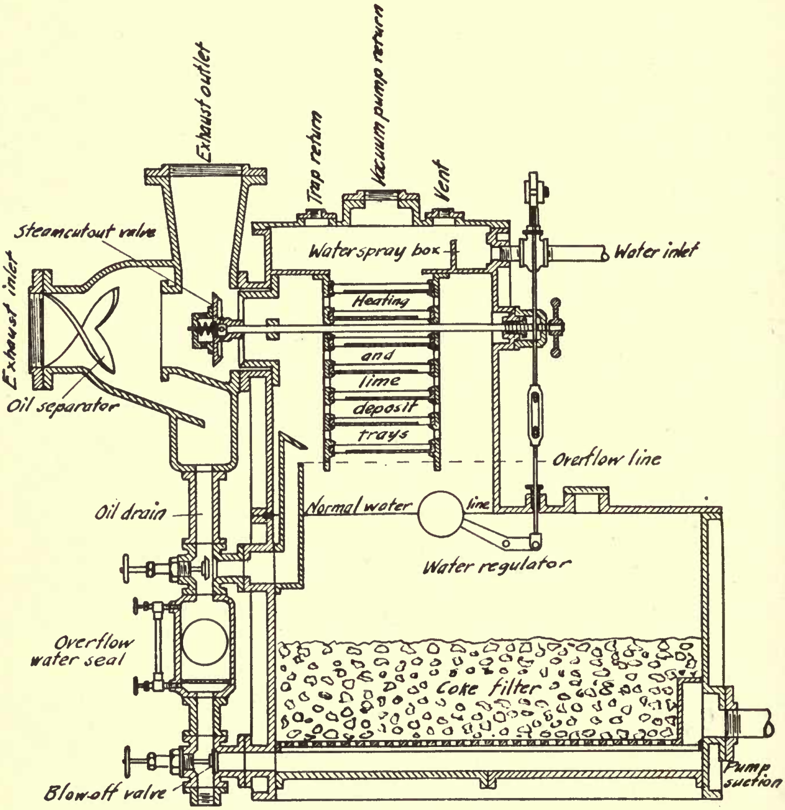


Fig. 34

The most common kinds of covering are made from asbestos or a mixture of asbestos and carbonate of magnesia. The magnesia used in pipe covering contains a great number of very small air cells, and therefore makes an excellent insulator. When the magnesia is used it is usually moulded into hollow cylinders together with enough asbestos fiber to give it strength. Another form of covering much used is made of several thicknesses of corrugated asbestos paper formed into hollow cylinders by winding on a mandrel.

72. Feedwater Heaters. — In most plants, all or some of the steam is exhausted at atmospheric pressure. If this steam is exhausted to the air all of its heat is wasted. Some of this heat may be used to heat the boiler feedwater by running the exhaust steam through a feedwater heater and extracting its heat of vaporization. There are two types of heater, the *open* and the *closed*.

In the open feedwater heater the steam comes in direct contact with the feedwater, which is made to flow over shallow pans, thus exposing a large area to the steam. The temperature of the water is thereby brought near to the boiling point. If the water is hard, a large part of the scale-forming materials will be deposited on the pans, which may be easily removed and cleaned. Figure 34 shows a common form of open heater. A skimmer is provided to remove the oil that comes in with the exhaust steam, and there is also a filter to purify the feedwater. The purpose of the open heater is thus seen to be twofold: to utilize the heat that would otherwise be wasted and to purify the water. In the closed type (Fig. 35), the steam and the feedwater do not come into direct contact. The steam is led through tubes around which the feedwater is forced to flow. If the water is very hard, the tubes are liable to collect scale, which hinders the operation of the heater.

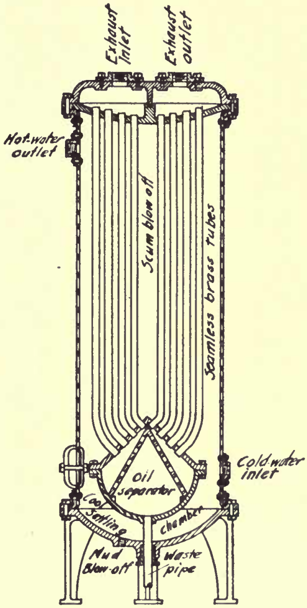


FIG. 35

73. Economizers. — In the ordinary steam plant, the flue gases pass up the stack at a temperature of about 500° F. This temperature usually will be higher than that of the steam and water in the boiler, since the latter get their heat from the gases. Moreover, the higher the steam pressure and its temperature, the hotter will be the flue gas. The tendency during the past few years has been to use higher pressures, which means a greater loss of heat up the stack than with low pressures.

In order to utilize a part of this heat that otherwise would be wasted, economizers are sometimes installed between the boiler and the stack. The economizer is simply an added heating surface in the form of water tubes about which the products of combustion pass on their way to the stack. At best the feedwater will be at a temperature of only 212° as it enters the boiler, if it has been heated with exhaust steam at atmospheric pressure. Considerable heat can be added before it reaches the boiling point when under high pressure. This heat is added in the economizer. The boiler feedwater is first pumped to the economizer, where it is heated to near the boiling point corresponding to boiler pressure, and it then passes on to the boiler.

In a common type of economizer, the heating surface is composed of vertical tubes through which the water flows and around which the hot gases pass. These tubes are kept clean from soot by scrapers that are continually moved up and down the tubes, by means of a small engine or electric motor. As the economizer depends for its action upon the extraction of heat from the burnt gases, it follows that the gases will be much cooled, and if natural draft be employed, they may be cooled enough to reduce the draft to such an extent that the efficiency of the whole plant may be lowered. If forced draft is used this objection does not hold to so great an extent. In any case, the economizer offers some resistance to the gases, with a consequent lowering of the draft. Whether or not an economizer will effect enough of a saving to pay for itself must be determined in each individual case. Economizers are often sold under a guarantee to add a certain percentage to the efficiency of the whole plant.

74. Condensers. — After steam has passed through an engine or turbine, it is often led to a condenser, in which a pressure considerably below that of the atmosphere is maintained. The process has several advantages which will be studied in more detail later. In general, the decreased back-pressure adds to the efficiency and to the capacity of the engine or turbine to an extent more than sufficient to pay for the additional cost of the condenser, provided plenty of cool water is available for cooling the condenser so as to condense the steam. Moreover, with a surface condenser, the condensed steam is led back to the boiler and is thus kept free from scale-forming materials. This last factor is of

great importance where the available feedwater is poor. Boilers can be operated far beyond their rated capacity if they are kept free from scale. Hence the saving in boilers and in their upkeep may also go a long way toward paying for the condenser.

There are two types of condensers, one in which the cooling or circulating water is kept separate from the steam, as in the closed feedwater heater, and one in which the water (called injection water) is mixed with the steam, as in the open feedwater heater. The former is called a ***surface condenser,*** and the latter a ***jet condenser.***

75. The Surface Condenser. — Figure 36 shows the construction of a surface condenser. The exhaust steam enters at the

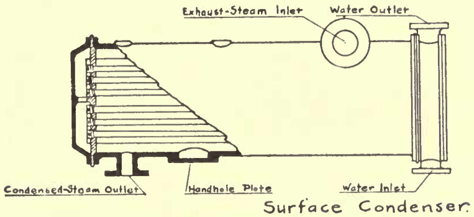


FIG. 36

top of the shell, passes around the tubes, and, after being condensed, is pumped out from the bottom of the shell. The tubes are usually made of thin brass or of special metal, and extend between two tube sheets. At the outer side of these tube sheets and within the tubes is the water space. The circulating water is pumped in at the opening shown in Fig. 36, flows through the lower half of the tubes to the other end of the condenser, and then flows back through the top tubes and out. This arrangement is called a ***two-pass condenser.*** In some smaller types, the water enters at one end and flows out at the other; these are called ***one-pass condensers.***

Occasionally three passes are made, but this type is not general. In the design of a condenser, care must be taken that the steam, upon entering, is directed over the entire surface of the tubes and that no air pockets may be formed. The condensed steam should leave that part of the condenser where the circulating water is coldest. Packed joints are used between the tubes and the sheet.

Since water will absorb and dissolve air when they come into contact, some air will be taken into the boilers with the feedwater and will pass over with the steam to the engine. Air also may leak into the steam through the stuffing boxes of the engine or turbine when run condensing. This air would soon clog the condenser and prevent condensation of the steam if it were not removed. The air is pumped out either with the condensed steam by means of a *wet-air pump* or else separately by means of a *dry-air pump.*

The circulating water is pumped through the condenser by the circulating pump. To maintain a high vacuum, the circulating water must be at a low temperature when it leaves the condenser. This means that each pound of circulating water can absorb only a few B.t.u. Each pound of the steam that condenses gives up to the water something like a thousand B.t.u. It therefore is evident that a large volume of circulating water must be used. As the pressure to be pumped against is small, a centrifugal pump is commonly employed for forcing the circulating water through the condenser. Air pumps are made both of the reciprocating type and of the rotary type. The former are more common. The design of the air pump is a rather difficult problem, since the air is very rare at a high vacuum, so that if the pump has much clearance it will fail to maintain this vacuum.

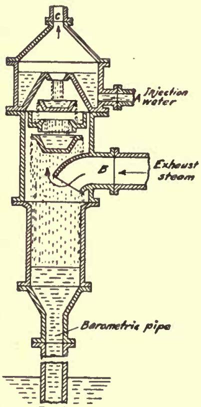


FIG. 37

76. The Jet Condenser. — As stated before, the steam and water mix in the jet condenser. This mixture of condensed steam, injection water, and the air contained in the steam and water, may be pumped out by a wet-air pump.

However, the water is often allowed to run out of the condenser by gravity through a vertical pipe thirty feet or more in length that has its lower end submerged. The air is pumped out by a dry-air pump. This arrangement is commonly called a siphon or barometric condenser.

Figure 37 shows in section this latter type of jet condenser. The injection water enters at A and runs over the edges of trays, thus exposing a large surface to the steam which enters at B. The air pump sucks the air out at the top through the pipe C.

In this type, the flow of the steam, until it is condensed, is with the air and opposite to that of the water. Such a condenser therefore is called a counter-current condenser.

Another type of jet condenser in which the air pump is dispensed with is shown in Fig. 38. The air is carried along with the condensed steam and the injection water. This is due to the high velocity of the water as it passes out of the constricted opening *A*. This is called the injector or ejector type. It usually is furnished with a barometric tube, as in the preceding type.

The jet condenser is more compact and less expensive than the surface condenser. If it is well made and equipped with a good air pump, it will give a very high vacuum, but it mixes fresh water with the condensed steam, and this may cause scale if the mixture is used for the boiler feed.

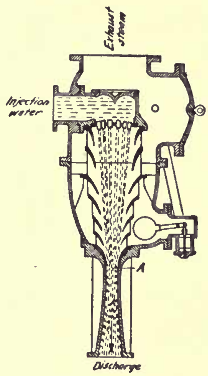


Fig. 38

77. Cooling of Circulating Water. — Since a large amount of cold water is costly in some locations, it is sometimes necessary to cool the circulating water so that it may be used over and over. This may be done by running it into a pond where the natural evaporation from the surface will cool it. If the space for a large pond is not available, the evaporation may be increased by the use of spray nozzles which break the water up into a fine spray, thereby exposing a large surface for evaporation. Another method for the rapid cooling of the circulating or injection water is the use of cooling towers. The water is allowed to trickle down over a lattice or other surface, thus exposing a large surface for evaporation. Air is either passed up through the tower by natural draft, or blown through by means of fans. Where large bodies of cold water, such as rivers or lakes, are available, the circulating water is drawn from them and is thrown away after being used. If the water is taken from lakes or rivers, it is often necessary to pass it through a screen to remove such material as weeds, drift, and fish. These screens require cleaning periodically and are sometimes made in the form of an endless chain, so that while some sections are being cleansed others may be in use.

CHAPTER VI

THE STEAM ENGINE

78. History. — In 1698 THOMAS SAVERY produced the first steam engine that proved to be commercially successful. It was used for pumping water. The engine consisted of two egg-shaped vessels, each of which connected with the supply of water to be pumped and to a steam boiler. In its operation steam was admitted to one of the vessels, and when it was full, connection with the boiler was cut off. Cold water was then sprayed on the outer surface of the vessel, causing the steam inside to condense and to form a partial vacuum. This vacuum opened a valve in the pipe leading to the well and sucked water into the vessel. Steam was then turned on again, and the pressure forced the water out of the vessel through the delivery pipe. When the water had all been forced out, the process described above was repeated. The engine was operated so that while one cylinder was forcing, the other was sucking water. This engine is the same in principle as the modern pulsometer.

Since the steam in the Savery engine came in direct contact with the water during the forcing stroke, there was much loss of steam by useless condensation. DENIS PAPIN, in 1705, made an improvement on the Savery engine by making the steam vessel of cylindrical shape and separating the steam and water by a floating piston, thereby preventing a part of the unnecessary condensation.

About 1711, there came into use a machine that was known as the Newcomen engine. THOMAS NEWCOMEN, with the aid of JOHN CALLEY, and with certain ideas from Papin, made his engine with a vertical cylinder into which a piston was fitted from the upper end. The cylinder was placed directly above the boiler and connected with it. Steam was admitted to the cylinder by the opening of a valve placed between the boiler and the cylinder. The piston was connected to a pump through a walking beam, one end of the walking beam connecting to the pump rod and the other to the piston rod. The beam was so counterbalanced that it took but little steam pressure to force the piston up. Steam was generated at about atmospheric pressure, and as the piston

moved up the steam valve was opened and steam filled the space beneath it. When at the top of its stroke, water was sprayed into the cylinder, condensing the steam and forming a partial vacuum. This vacuum under the piston allowed the atmospheric pressure from above to force the piston down.

While the Newcomen engine was an improvement over the Savery engine, it was very wasteful of steam because the cylinder was cooled by the spray of water on each downward stroke. Much condensation of steam occurred in heating up the cylinder walls on each upward stroke. While repairing one of these Newcomen engines, James Watt conceived ways in which it might be improved. Patents covering these ideas were granted him in 1769. Watt's chief aim was to keep the cylinder walls as hot as the incoming steam at all times and thereby prevent the initial condensation that rendered the older engine so inefficient. This high cylinder temperature was to be maintained, first, by condensing the steam in a vessel away from the cylinder, and second, by a steam jacket placed around the cylinder walls. Lagging was also to be placed around the outside of the cylinder to keep down the heat lost to the outside air. In the previous engines, the piston was kept tight by a kind of water seal on top of the piston. Watt used fibrous packing and tallow to keep the piston tight and saved heat that previously was lost to the water above the piston. In the operation of his engine Watt found it necessary to remove the air from the condenser and so he equipped the condensers with air pumps.

While it is seen that Watt is not the inventor of the steam engine, yet it must be admitted that he did more to advance its development than any other one man. Up to the time of Watt, the steam engine was used almost exclusively for pumping water in collieries, but he applied it to the driving of other forms of machinery. After many hardships and discouragements, Watt at last was able to produce his engine in large numbers. The engine became increasingly popular, and we may say that the era of our present industrial development started at the time of James Watt. In applications for his patents Watt advocated the use of high-pressure steam from which work could be obtained by using it expansively, but in the actual construction of his engines he never used pressures much above that of the atmosphere.

Since the time of Watt, various improvements have been made

in the steam engine. The mechanical construction has been bettered, the valve mechanism improved, compounding adopted, and the steam pressures greatly increased. While its thermal efficiency may not be as high as some forms of internal-combustion engines, the steam engine is very reliable.

79. The Plain Slide-valve Engine. — Figure 39 shows in vertical and horizontal sections the parts of a simple steam engine. Its action is as follows: Steam comes from the boiler through the steam-pipe, and after passing the throttle valve, enters the steam chest. The valve, driven by an eccentric on the shaft, moves backward and forward on the valve seat, uncovering alternately the two steam ports. When a steam port is uncovered by the valve, the steam flows through the port into the cylinder and by its pressure moves the piston in the cylinder. In Fig. 39, the left steam port is shown partly open and the steam is then pushing the piston to the right. At the same time that the steam is forcing the piston to the right, the valve has uncovered the right port, so that the steam on the right of the piston may escape to the exhaust pipe.

This motion of the piston is transmitted by the piston rod to the cross-head and from this through the wrist pin and the connecting rod to the crank. The reciprocating motion of the piston is transformed by the connecting rod and crank into rotary motion of the shaft. The power generated in the cylinder usually is taken from the shaft by a belt on the flywheel or by an electric generator coupled to the shaft. The valve is made to move so that when steam is being admitted to one end of the cylinder it is being exhausted from the other. As the incoming steam is at a much higher pressure than the exhaust, there is a resultant force pushing the piston in the direction of the outgoing steam.

That end of the cylinder farthest from the crank is called the ***head-end***, and the end nearest the crank the ***crank-end***. With the piston at the extreme left of its travel, the crank will be in a direct line between the cylinder and the shaft. While the force on the crank pin may be large, there is no turning effort. The crank is then said to be on ***head-end dead center***. With the piston at the extreme right of its travel the crank is on ***crank-end dead center***. When the engine is running, if the crank rises, as the piston leaves the head-end dead center (*i.e.*, if in Fig. 39

the crank moves in a clockwise direction), the engine is said to be ***running over.*** If the crank moves in the opposite direction, the engine is said to be ***running under.***

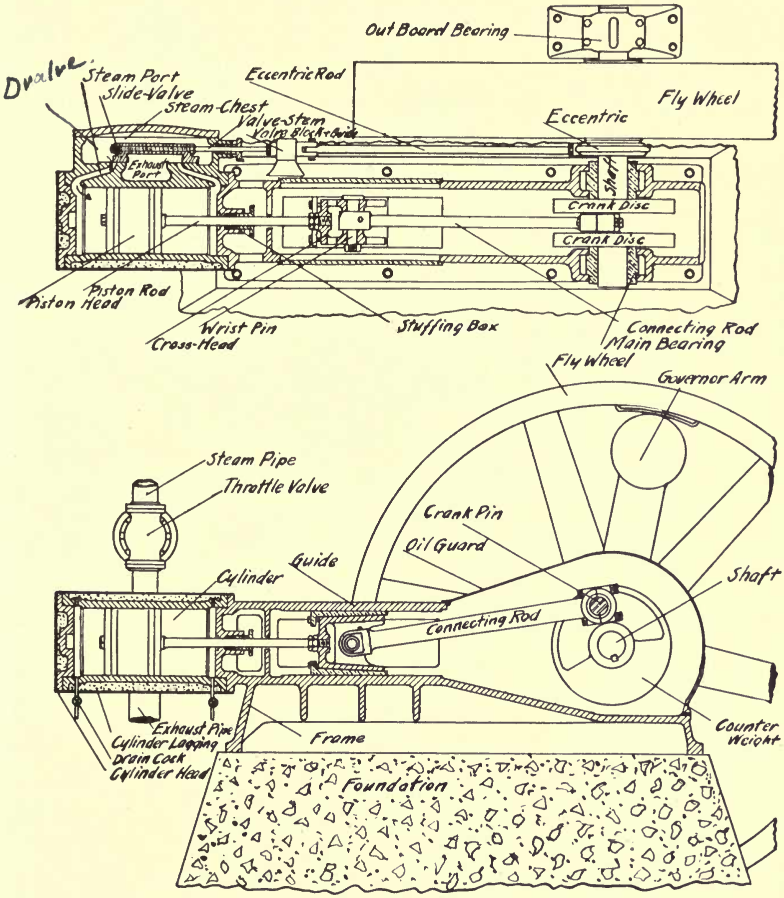


FIG. 39

The ***stroke*** of the engine is the distance the piston travels in half a revolution. It is equal to twice the length of the crank. On the ***head-end stroke,*** the piston moves from head-end to crank-end dead center, and the reverse motion takes place on the ***crank-end stroke.***

80. Parts of the Steam Engine. — CYLINDER. Steam-engine cylinders are made of cast iron and the bore is carefully machined. With proper lubrication, the surface exposed to wear acquires a high polish and the metal is worn away very slowly. The ports are cored in the casting and are not finished except along the edges at the valve seat. At each end of the cylinder the diameter is made slightly larger. This enlarged part is called the ***counter-bore.*** It should extend far enough so that the piston ring comes to its edge, in order that a shoulder may not be worn in the cylinder wall. The counterbore also serves a purpose when the cylinder has to be rebored, since the boring machine may be set on the counterbore, which will not be worn away, and thus the alignment need not be lost. On some larger engines used in marine service, a thin inner shell or liner is placed in the cylinder, so that it may be replaced without changing the cylinder when it becomes worn.

The cylinder head is bolted to the cylinder, and the joint is made steam-tight by means of a gasket or a ground joint. In the smallest engines the cylinder is cast as a part of the frame, but ordinarily it is cast separately and bolted to the frame.

In large engines where high efficiency is desired the cylinder may have a steam jacket. The heads may or may not be jacketed. In any case the cylinder is covered by some non-conductor of heat, called lagging. With proper lagging very little heat is lost by radiation.

Unless the exhaust valves are so placed that any water in the cylinder can drain to them, it is necessary to tap the bottom of the counterbore at each end of horizontal cylinders and place a drain cock there. These cocks are opened when the engine is warming up so that the condensed steam will not be caught when the piston comes to the end of the stroke. As water is incompressible, its presence in too large a quantity would cause the breaking or straining of some part. Sometimes automatic relief valves are placed at the ends of the cylinders to take care of any water that may get into the cylinders. Each end of the cylinder is tapped for the connection of an ***indicator.***

VALVES. The subject of valves will be taken up in detail in Chapter VIII.

PISTON. Pistons for stationary engines are made of cast iron. The piston is turned to a slightly smaller diameter than the

cylinder, and leakage of steam past it is prevented by the use of piston rings which fit into grooves cut around the piston. In the small sizes the rings are made in one piece slightly larger than the diameter of the cylinder. A piece is then cut out of each ring, and they are snapped into the groove in the piston. Because of their elasticity, they spring out and make contact with the cylinder walls. When worn they may be replaced by new rings. On larger pistons the rings are built up in sections and are pushed out against the cylinder wall by springs placed under them.

On small engines the pistons are cast in one piece, and usually are hollow, to make them as light as possible. In the larger sizes, they are built up of two or more pieces. In vertical engines and locomotives, the pistons sometimes are made dished to a slight extent. The dishing may be to add strength, to shorten the length of the engine a small amount, or to facilitate drainage.

Piston Rod. The piston rod connects the piston to the cross-head. It is made of steel, and the connection must be such that it will be always tight. If a little play is allowed, a bad knock develops that rapidly grows worse. In horizontal engines, a tail-rod sometimes extends from the piston out through the head-end cylinder head, and its outer end is carried by a slipper on a guide. This arrangement allows the weight of the piston to be carried by the slipper and the cross-head and lessens the wear on the piston and the cylinder.

Stuffing Box. The joint between the piston rod and crank-end cylinder head is made tight by a stuffing box. The packing used on low-pressure engines in this box may be fibrous, but with high steam pressure a metallic packing is commonly used. A good packing should keep the joint steam-tight and at the same time give but little friction on the rod. Wear in the cylinder, improper adjustment of the cross-head, poor alignment, or a pitted or scored rod, may cause excessive wear on the packing. It is then difficult to keep it steam-tight.

Cross-head. The cross-head with the *cross-head pin* or *wrist pin* forms the connection between the piston rod and the connecting rod. The cross-head is made to move in a straight line by guides on the frame of the engine. The two-guide type shown in Fig. 39 is the most common, although four-guide and one-

guide types are used occasionally. The slipper type also is often used; in this the cross-head takes the form of a slipper which slides on the flat surface of one guide. In all types the wear between the cross-head and guides is taken up by the adjustment of the wedge-shaped slippers or by the use of shims. In a few steam engines and most gas engines a trunk piston is used. In this type the piston itself acts as cross-head and carries the wrist pin. With this latter arrangement, the engine is single acting, *i.e.* the steam acts only on the head of the cylinder and on the face of the piston.

CONNECTING ROD. The connecting rod connects the wrist pin to the crank pin. It is alternately in compression and tension, and usually is made of steel. On high-speed engines, considerable bending stress may be developed in the connecting rod on account of its fling, and hence its cross-section is usually rectangular or I-section. On slow-speed engines this bending stress is small, and the rod is made circular in section. Brass or other metal bearing-pieces called ***brasses*** are used at the bearing points on the pins. These brasses are often babbited. As wear occurs adjustment must be made to take it up. If the rod is shortened in taking up the wear, the end on which such adjustment is made is said to have an ***open stub-end***. If the rod is lengthened by an adjustment at one end, that end is called a ***closed stub-end***. In Fig. 39, the end at the crank pin is of the marine stub type, which is used on all center-crank engines. In this last type the wear is taken up by removing liners, and in the former types usually by means of wedges which move the brasses.

CRANK. The crank pin is made of steel, and it may be a part of the same forging as the crank and shaft, or it may be set into crank discs which are keyed to the shaft. When the pin is placed between two crank discs, as in Fig. 39, we have a ***center-crank engine;*** when it overhangs one crank disc, we have a ***side-crank engine***.

COUNTERBALANCE. The crank pin and half of the connecting rod usually are considered as rotating parts and must be counterbalanced to make the engine run smoothly. Furthermore the piston, piston rod, cross-head, and half of the connecting rod have a reciprocating motion. It takes a large force to start and stop them on each stroke. Unless they are counterbalanced the whole

engine will vibrate on the foundation. While it is impossible to counterbalance exactly both the rotating and the reciprocating parts at the same time, yet it can be partially done. The proper sized counterbalance or counterweight (Fig. 39), sometimes made of lead but usually of iron, is put on to give smooth running.

SHAFT. Engine shafts usually are made of steel. As explained above, they are either forged to make the crank and crank pin integral parts of the shaft, or else the crank is keyed to the shaft. In addition to the key, a shrunk fit sometimes is used, or the crank disc may be pressed on by hydraulic pressure.

BEARINGS. With a side-crank engine there is one main bearing, and with a center-crank engine, there are two. The weight of the flywheel may be carried partly by an outer bearing called an ***outboard bearing.*** There will be wear on the main bearing in a vertical direction on account of the weight of the flywheel and of the rotating parts, and there will be wear in a horizontal direction on account of the thrust from the piston. Many main bearings are made up of four parts, the cap, the bottom part which takes the vertical wear, and two side-pieces which take the horizontal wear. The latter are called quarter-boxes. These parts may be adjusted separately.

FLYWHEEL. The turning moment on the crank varies at different parts of the stroke. At dead center it is zero. In order to keep the shaft turning at approximately the same speed at all times in the revolution, a flywheel is put on the shaft. This acts to store up and give out energy at the proper times, thereby keeping the angular velocity approximately uniform. The flywheel may carry the belt or there may be a separate belt wheel in addition to the flywheel, in which case the latter is sometimes called a balance wheel. The flywheel commonly is made of cast iron. In the smaller sizes it is cast in one piece, but in the larger sizes, it is cast in sections. Great care must be taken in its manufacture, since a crack may cause a disastrous accident.

ECCENTRIC. An eccentric is placed on the shaft or governor arm to drive the valve. It is encircled by the eccentric strap, which is connected to the valve by means of the eccentric rod and valve stem. This is really a substitute for a crank and connecting rod and gives to the valve a motion similar to that of the piston.

Frame. The frame is made of cast iron in stationary engines, and the better engines have the heavier frames. The greater the weight of frame the more smoothly the engine will run, other things being equal. On some small high-speed engines there is a cast-iron *sub-base* placed between the frame and the foundation. Frames are given different names according to their shape and the cylinder arrangement.

Foundation. Foundations usually are made of brick or concrete. The latter is now the more common. The frame is fastened to the foundation by anchor bolts. The foundation should be quite massive and should rest on soil that is firm enough to carry the weight of the engine and foundation without settling.

81. Piston Displacement. — The volume the piston displaces in moving from one dead center to the other is called the ***piston displacement.*** It is commonly expressed in *cubic feet.* The head-end piston displacement is equal to the length of stroke in feet times the area of the piston in square feet. The crank-end piston displacement is the area of the piston minus the area of the cross-section of the piston rod, times the stroke.

The ***size of an engine*** is given in *inches,* the diameter of the cylinder bore first and the length of the stroke second, *e.g.* an $18'' \times 24''$ engine is one whose cylinder is $18''$ in internal diameter and whose stroke is $24''$. The ***size of a compound engine*** is given by the diameter of the high-pressure cylinder, the diameter of the low-pressure cylinder, and the stroke, as $10'' \times 18'' \times 24''$. If the volume is calculated in cubic inches, remember to change tc cubic feet in giving the piston displacement.

82. Clearance. — When the piston is at the extreme end of its travel, there will be some volume back of it, because it is necessary to have a little space in which to take up the wear on the connecting rod brasses and to allow for unequal expansion of parts as the engine heats up, and because of the space in the ports. This volume, the larger part of which is often the volume of the ports, is called ***clearance.*** Clearance is expressed as a percentage of the piston displacement. Thus, when we say that the head-end clearance of an engine is 4.5 percent, we mean the volume back of the piston when the engine is on head-end dead center is .045 times its head-end piston displacement.

To determine the clearance of an engine, first place the engine

on the dead center of the end for which the clearance is to be measured. Second, disconnect the valve from the eccentric rod, move it so that the port for that end is closed, and block it up in that position. It is necessary to disconnect the valve because the port is usually open a small amount when the engine is on dead center. Third, pour water into the opening for attaching the indicator cock until the clearance space is full of water. If the valve is placed at the top of the cylinder, as is the case with horizontal Corliss engines, remove the valve and pour the water into the port. Having recorded the amount of water poured in, and having made correction for leakage, the volume of water necessary to fill the clearance space is computed. This volume divided by the piston displacement for that end gives the clearance. A rough method of computing clearance from the indicator diagram is given in § 89.

83. Steam Back of Piston during Stroke. — The weight of the dry steam back of the piston may be computed for any percent of the stroke if we know the clearance, the size of the engine, and the steam pressure at that percent of the stroke. Add the percent of the stroke to the percent of clearance. Multiply this by the piston displacement, and divide by 100. The result is the volume of steam back of the piston at the given percent of stroke. By means of the indicator, the pressure may be determined for the same position. The density of steam may be read from the steam tables for that particular pressure. The product of this density and the volume back of the piston gives the weight of the steam there.

Example. What is the weight of dry steam back of the piston at 27% of the crank-end stroke of a $14'' \times 16''$ engine? The engine has a 2″ rod and the crank-end clearance is 6.3%. The crank-end indicator card is at hand.

Solution. Measure the pressure from the card at 27% of the stroke. Suppose this pressure is 115 pounds gage, and the atmospheric pressure is 14.5 pounds per square inch. The absolute pressure is then $115 + 14.5 = 129.5$ pounds per square inch. The density of dry saturated steam at this pressure is, from the steam tables, .2887 The piston displacement is seen to be $16(\pi 7^2 - \pi)/1728 = 1.398$ and the volume back of the piston is therefore equal to $(.27 + .063) \times 1.398 = .466$ cubic feet. The weight of dry steam is then $.2888 \times .466 = .1342$ pound.

84. The Indicator and Its Purposes. — The ***steam-engine indicator*** was first used by James Watt, who invented it. Since his time it has been perfected and is now very extensively used.

The indicator records on paper a line showing the relation between the pressure in the cylinder and the movement of the piston. The diagram or card produced is of value in setting the valves, in computing the horsepower developed in the cylinder, and in making analyses of the operation of the engine. A description of the mechanism of the indicator will not be given here. When we speak of the ***scale of spring*** of an indicator, we do not mean the actual scale of the spring used in the indicator, but rather the relation between the movement of the pencil on the indicator diagram and the pressure that causes it. That is, a 60-pound indicator spring is one that gives a pencil movement of one inch for each 60 pounds per square inch increment of pressure in the cylinder.

The heavy lines of Fig. 40 show a representative diagram as it comes from the indicator. It is seen that the diagram is a closed irregular curve with a straight line underneath. The straight line is the atmospheric-pressure line, which is used for reference in measuring pressures. The ordinates of the points on the curve show to some scale the pressures in the cylinder, and the abscissas the movement of the piston. Starting at the upper left-hand corner of Fig. 40, we have the pressure back of the piston when it is on dead center. As the piston moves forward, the pressure changes as shown by the upper curved line of the diagram. It is seen that the pressure drops but little for the first part of the stroke, and later more rapidly, until at the end of the stroke it is nearly down to that of the atmosphere. On the backward stroke, the pressure remains nearly constant until near the end, when it rapidly rises to the initial point.

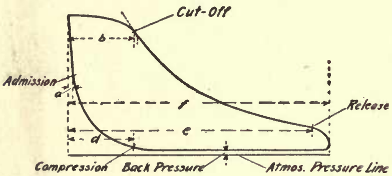


FIG. 40

85. Events of Stroke. — There are four events of the stroke. 1. ***Admission*** occurs when the valve uncovers the port and allows the steam to enter the cylinder. 2. ***Cut-off*** takes place when the valve closes the port and prevents any more steam from entering. 3. At ***release,*** the valve uncovers the port and allows the steam to escape to the exhaust. 4. ***Compression*** occurs when the valve again closes the port and prevents any more steam from leaving the cylinder for the remainder of the stroke.

It is seen that the steam enters the cylinder from admission to cut-off, and that the steam thus let in expands and does work on the piston from cut-off to release. From release to compression, the used steam is being exhausted from the cylinder. The steam that is caught when the exhaust port closes is compressed into the clearance space during the time from compression to admission.

86. Location of Events on Diagram. — After a little practice the events may be quite accurately located on the ordinary indicator diagram or card. In Fig. 40 it is seen that the upper line drops down and that there is a point of inflection in this curve. This is the point of cut-off. This point of inflection is easily detected by drawing a continuation of the two curves as shown by the dotted lines at the point of cut-off. Release occurs at the next point of inflection; it may be located in a manner similar to that in which we located the cut-off. At compression there is no point of inflection; therefore its proper location is difficult. The exhaust valve ordinarily closes rather slowly, and the passageway for steam being small when it is nearly shut, the pressure may start to rise even before the valve is completely closed. A common error is that of taking the point of compression too low on the compression curve. Admission occurs where the compression curve stops and the straight line starts.

To determine the percentage of stroke at the different events, draw the two end-ordinates. The distance between these, f in Fig. 40, represents the length of stroke to some scale. The distances from the left end-ordinate to the events shown by a, b, e, and d, divided by the distance f, give the ratios of the stroke at these events, and the percentages of stroke are those ratios multiplied by 100.

87. Equation of Expansion and Compression Curves. — There is a definite relation between pressure and volume during expansion and compression. The equation $pv^n = p_1v_1^n = p_2v_2^n$ expresses this relation, where p is the absolute pressure on these curves, v is the volume back of the piston (including the clearance space), and n is some constant exponent for each individual curve. An analysis of many cards shows that n is sometimes a little less than 1 and sometimes a little larger than 1. For rough calculations, it may be considered equal to 1. With this assumption, the equation of the expansion and compression curves is $pv = C$.

This is the equation of the equilateral hyperbola. Certain geometric facts about this curve are of importance to us. In Fig. 41, let us choose any two points on an equilateral hyperbola: E, whose coördinates are (p_1, v_1), and H, whose coördinates are (p_2, v_2). We shall show that the line of the diagonal CA of the rectangle $ECHA$ drawn through these points, passes through the origin. In the triangles OAB and OCD, $OB = v_1$, $BA = p_2$, $OD = v_2$, and $DC = p_1$. Since their sides are parallel, these two triangles are similar. Hence we have

$$\frac{OB}{OD} = \frac{AB}{CD}, \text{ or } \frac{v_1}{v_2} = \frac{p_2}{p_1}, \text{ or } p_1v_1 = p_2v_2,$$

which satisfies the equation of the curve.

It follows that we can construct the entire curve if one point E on it is given. We can locate other points on it in the following manner. Through the point E draw horizontal and vertical lines. Choose some point C on the horizontal line, and draw the line OC through the origin. Drop a vertical line from C, and draw a horizontal line through A, the point where the line OC cuts the vertical line through E. The intersection H of the vertical line through C and the horizontal line through A is a point on the curve. Other points may be found in the same manner.

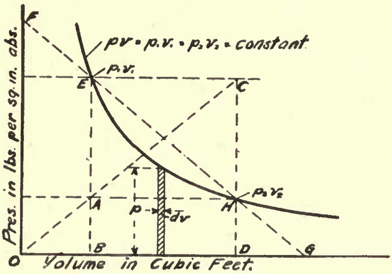


FIG. 41

Another geometric fact of value is that the length of FE is equal to HG on a line drawn through the points E and H. This may be proved readily from Fig. 41. The area under the curve from E to H, *i.e.* $BEHD$, may be determined by integration. The increment of this area has dimensions p and dv, and dA is equal to pdv. Hence the total area is

$$A = \int dA = \int_{v_1}^{v_2} pdv,$$

but since $p_1v_1 = p_2v_2 = pv$, we have

$$p = \frac{p_1v_1}{v},$$

and

$$A = p_1v_1\int_{v_1}^{v_2}\frac{dv}{v} = p_1v_1 \log_e\left(\frac{v_2}{v_1}\right).$$

88. Hypothetical Indicator Diagram. — Diagrams are sometimes constructed on the hypotheses that the expansion and compression curves are equilateral hyperbolas, that the pressure during the admission of steam from the end of the stroke to cut-off is constant, and that the back pressure is constant up to the point of compression. Such a diagram will be called the ***hypothetical diagram.*** This diagram is also sometimes called the *theoretical*, or *ideal*, or *conventional* diagram.

The construction of such a diagram, shown in Fig. 42, is carried out as follows. First choose a suitable length f for the diagram, and draw in the atmospheric-pressure line. Next choose a scale of pressures, and draw the volume axis at a distance c, the atmospheric pressure, below the atmospheric line. Draw the pressure axis at a distance from the point F of the diagram equal to the ratio of clearance times the length f of the diagram. Measure up from the atmospheric-pressure line the initial steam pressure, and draw the steam-admission line FA. The length of FA is equal to the ratio of cut-off times the length f. From the point A, construct an equilateral hyperbola as in § 87. The length e is equal to the ratio of release times f. From B, the point of release, draw a line to the end of the diagram at C. The distance h from C to the atmospheric-pressure line represents the back pressure. From C to D draw the back-pressure line parallel to the atmospheric line. From D, the point of compression, construct an equilateral hyperbola to the point E, whose distance a from the end of the diagram is the admission distance. Connect E and F by a straight line.

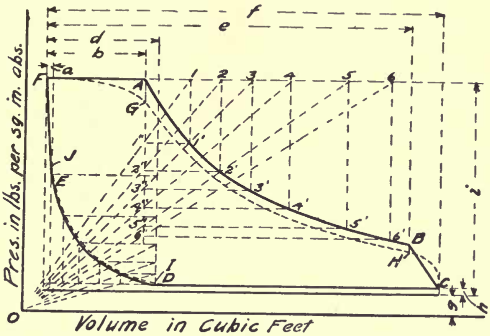


FIG. 42

The actual diagram may vary considerably from the diagram just constructed. The speed of the engine, the throttling of steam in the ports and by the valve, the condensation in the cylinder, etc., affect the form of the actual diagram, which may resemble the dotted diagram $FGHCIJF$ in Fig. 42. If the en-

gine exhausts into a condenser in which a vacuum is maintained, the back-pressure line will fall below the atmospheric-pressure line.

89. Determination of Clearance from Card. — Since it is often impossible to measure the clearance of an engine when a test is being made, a rather rough approximation sometimes is used. Suppose the indicator diagram is as shown in Fig. 43. Two points A and B are chosen on the compression curve. On these two points a rectangle is drawn, and the diagonal is extended until it cuts the volume axis, which is drawn below the atmospheric line at a distance equal to the barometric pressure. The point of intersection of this diagonal and the volume axis establishes the origin, and the pressure axis may be drawn. The distance c divided by the length of card gives the ratio of clearance. The pressure axis may also be located by drawing the other diagonal through A and B and laying off DA equal to BE. If the piston rings are not tight in the cylinder, or if the valve leaks steam, this method will not give even approximately correct results.

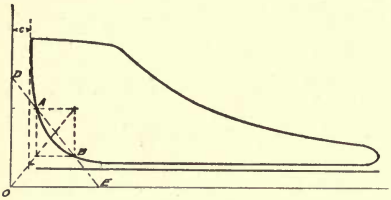


FIG. 43

90. Determination of the Mean Effective Pressure. — It has been mentioned that the indicator diagram shows the pressure at all points of the stroke. Since the total pressure on the piston times the distance the piston moves is the work done by the steam, we see that the indicator card is a work diagram.

In order to calculate the work done in the cylinder we must know the ***average effective pressure,*** or the ***mean effective pressure*** (***m. e. p.***) on the piston. In Fig. 44 when the piston is at G on the forward stroke, the steam pressure is a. When the piston is at G on the backward stroke, the steam pressure is b. During the forward stroke the steam is working on the piston, but on the backward stroke the piston is doing work on the steam.

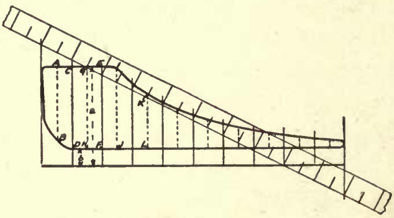


FIG. 44

Hence the effective pressure for the two strokes at G is $a-b$, which is shown by the dotted ordinate GH.

To get the *average*, or *mean*, of these effective pressures for the whole card, we may proceed as follows. Draw in the end-ordinates of the diagram. By means of a scale or the edge of a ruled piece of paper, divide the length of the card into a number of divisions of equal length. The number of divisions should be more than eight, and need not be more than fifteen. Through these division points, draw in ordinates as shown by the full lines. The diagram is thereby cut into a number of strips of equal width. The average height of each strip is about equal to the dotted ordinate located midway between the solid lines,

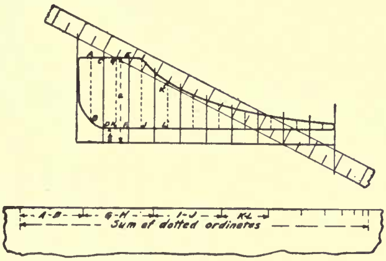


FIG. 45

i.e. the average height of the strip $CEFD$ is GH. While this may not be true in every case, the fact that some parts of the diagram are concave while other parts are convex will tend to neutralize the error when the dotted ordinates are averaged. Next add the lengths of the dotted ordinates. This is best done by laying a strip of paper on the diagram and marking the different ordinates directly on the edge of the strip. Figure 45 shows the strip of paper with the dotted ordinates added graphically. The average of the ordinates is this sum divided by their number. This average height of card, multiplied by the scale of spring, gives the mean effective pressure (m. e. p.).

If the m. e. p. of many cards is to be found, it may be quicker to use a polar planimeter to get the area of the card. This area divided by the length gives the mean height. This mean height multiplied by the scale of spring gives the m. e. p. This method is usually no more accurate than the former, when the former is done with reasonable care.

91. Indicated Horsepower. — The mean effective pressure (m. e. p.) in pounds per square inch times the area of the piston in square inches gives the ***total average effective force*** exerted by the steam on the piston during the forward and backward stroke, or for one complete revolution of the crank. The ***work done*** by this force is equal to the force times the distance the piston moves. It must be remembered that the effective force on the piston was obtained by taking the difference of pressures during the forward and backward strokes. With this in mind, we see that *the work done on the piston per revolution is equal to the m. e. p. times the area of the piston, times the length of the stroke.* If the length of stroke is expressed in feet, the result will be in foot-pounds.

If N denotes the revolutions per minute (r. p. m.), L the length of stroke in feet, P the m. e. p. in pounds per square inch, and A the area of piston in square inches, *the foot-pounds of work done per minute is equal to $PLAN$. The* ***horsepower of the engine*** *is then $PLAN/33000$.* Since this result is obtained by means of the indicator, it is called the ***indicated horsepower*** (***i. hp.***).

If the engine is double-acting, the i. hp. for the crank-end is found in a similar manner, taking the m. e. p. from the crank-end card and using the area of the piston on the crank-end, which will be less than that for head-end because the piston rod occupies some of the area of the piston. For very rough work, the average of the m. e. p. for the two ends is sometimes taken and the area of the piston rod is neglected; then we have, approximately,

$$\textit{total indicated horsepower} = \frac{2\,PLAN}{33000}.$$

Example. The head-end m. e. p. is 42.6 and the crank-end m. e. p. is 45.1 pounds per square inch in a $12'' \times 18''$ engine running at 220 r. p. m. The diameter of the piston rod is two inches. What is the indicated horsepower?

Solution. The area of a $12''$ circle is 113.1 square inches and of a $2''$ circle is 3.1 square inches. The area of the head-end of the piston is then 113.1 square inches and of the crank-end $113.1 - 3.1 = 110.0$ square inches. The stroke is $18''$, or $1.5'$; hence we have

$$\text{head-end i. hp.} = \frac{42.6 \times 1.5 \times 113.1 \times 220}{33000} = 48.2 \text{ hp.,}$$

and

$$\text{crank-end i. hp.} = \frac{45.1 \times 1.5 \times 110.0 \times 220}{33000} = 49.7 \text{ hp.}$$

whence the total i. hp. is $48.2 + 49.7 = 97.9$ hp.

92. Brake Horsepower. — It is usually not difficult to take cards from an engine and to compute the i. hp. from them. This does not give the horsepower that the engine is actually delivering, of course, but that which is developed in the cylinder. In testing an engine that is in actual use, it is often impossible to measure its actual output without considerable trouble and expense. Under such conditions one must be content with the i. hp. If the engine is not too large and conditions will permit, the actual power delivered by the engine is often measured. If it is direct-connected to an electric generator and the losses in the generator are known, this is fairly simple. If not, some form of dynamometer may be used. The most common means of measuring the delivered horsepower is by a brake on the flywheel.

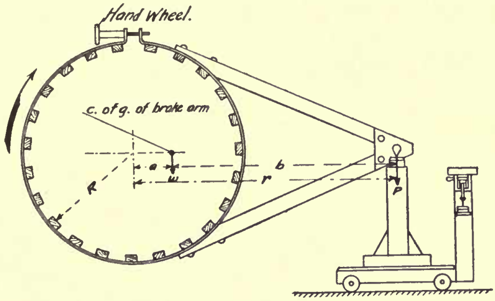


FIG. 46

If a number of tests are made, the ***Prony brake*** is generally used. For an occasional test a rope brake may answer the purpose.

Figure 46 shows a common form of Prony brake. Two or more bands of strap iron with wooden blocks fastened to them are put around the circumference of the flywheel or the belt wheel. The tension in the band is regulated by means of a hand-wheel shown at the top of the figure. Two arms are fastened to the band which at the right end in the figure carry a knife-edge that rests on a block placed on a platform scales. The flywheel rotates in the direction of the arrow, and the friction between the blocks and the wheel causes a pressure on the scales.

When the engine is not running, there will be some pressure on the scales due to the weight of the brake-arm. This may be determined by weighing the brake before putting it on the wheel and balancing it so that its center of gravity is determined. If

the weight of the brake is w and its center of gravity is located at a distance b from the knife-edge and at a distance a from the center of the wheel, we have, taking moments about the center of the wheel,

$$wa = \text{component of weight on scales} \times (a+b).$$

From this we can compute the effect of the weight of the brake arm on the scales. This component of weight must be subtracted from the pressure on the scales when the engine is running. If we denote by P the net pressure on the scales, which is due to the friction between the blocks and the wheel with the engine running, we may compute the power being absorbed by the brake as follows. The work absorbed by the brake per revolution is equal to $P \times 2\pi r$, in which r is the horizontal distance from the center of the wheel to the knife-edge and is known as the radius of the brake-arm. If the engine is running N revolutions per minute, the work absorbed per minute is $2\pi r P N$, and the horsepower absorbed is $2\pi r P N/33000$. This is called the ***brake horsepower*** (***b. hp.***). The radius of the brake wheel R does not enter into the computation of the b. hp. To prevent the burning of the wooden blocks, the wheel is kept cool by putting water inside the rim which evaporates and carries off the heat.

Example. During the test of an engine, the r. p. m. was 220. The pressure of the brake arm on the scales was 318 pounds (Fig. 46). The weight of the entire brake is 118 pounds, and its center of gravity (c. of g.) is 6.05 feet from the knife-edge. The radius of the brake-arm, r, is 7.16 feet. Find the brake horsepower.

Solution. If the c. of g. of the brake is 6.05 feet from the knife-edge, it is $7.16 - 6.05 = 1.11$ feet from the center of the wheel when the c of g. is in line between the knife-edge and the center of the wheel. That part of the weight of the brake supported by the scales is $118 \times 1.11/7.16 = 18.4$ pounds. The net pressure on the scales then equals $318 - 18.4 = 299.6$ pounds. The brake horsepower $= 2\pi rPN/33000 = 2\pi \times 7.16 \times 299.6 \times 220/33000 = 89.7$ hp.

93. Mechanical Efficiency. — The ratio of the brake horsepower to the indicated horsepower is called the ***mechanical efficiency.*** It is usually expressed in the form of a percentage. The difference between the indicated horsepower and the brake horsepower is the ***frictional horsepower.***

Example. If the i. hp. of an engine is 97.9, and the b. hp. is 89.7 hp. find the mechanical efficiency and the frictional horsepower.

Solution. The mechanical efficiency $= 89.7/97.9 = .916 = 91.6\%$. The frictional horsepower $= 97.9 - 89.7 = 8.2$ hp. Hence the frictional horsepower $= 8.2/97.9 = 8.4\%$ of the indicated horsepower.

94. Thermal Efficiency. — In general the efficiency of a machine is the ratio of the out-put to the in-put. In the steam engine heat is put in and mechanical work is taken out. ***The thermal efficiency of the steam engine is the ratio of the work got out to the work equivalent of the heat put in.*** The thermal efficiency may be based on either the i. hp. or the b. hp. The latter is called the ***overall efficiency,*** and is equal to the thermal efficiency based on i. hp. times the mechanical efficiency.

A common but approximate way of stating the efficiency of a steam engine is to give the weight of dry steam consumed per hour per horsepower. This may be based on either the i. hp. or the b. hp. It is not an exact way of stating the efficiency because steam may contain different amounts of heat, depending on pressures and superheat. Efficiency may also be expressed in terms of B.t.u. per minute per horsepower.

In computing the B.t.u. given to the engine in a unit of time, proceed as follows. The weight of dry steam is found by deducting the weight of moisture in the steam from the amount of wet steam used. The heat in this moisture is not charged to the engine, since it is not possible for the engine to extract work from it. From the steam tables find the total heat in a pound of dry steam, or the total heat in a pound of superheated steam if superheated steam is used. From this total heat per pound, subtract the heat of the liquid at the pressure of the exhaust. The reason for subtracting the heat of the liquid at the pressure of the exhaust is that although the engine has used the steam, the heat of the liquid can be saved by feeding the condensed steam back to the boiler, which is often done. Whether it is done or not, it is not fair to charge this heat to the engine. Multiply the amount of heat in a pound of dry steam that is charged to the engine by the weight of dry steam used in unit time. The result gives the B.t.u. upon which the efficiency is computed.

Example. An engine during a test developed 97.9 i. hp. and 89.7 b. hp. The engine used 3060 pounds of 97% quality steam per hour. The steam pressure was 125 pounds gage, and the engine exhausted to the atmosphere. The barometer reading was 29.3 inches. Find the thermal efficiency of the engine.

Solution. The weight of dry steam used per hour is $3060 \times .97 = 2970$ pounds. 125 pounds gage pressure $= 125 + 14.4 = 139.4$ pounds per square inch absolute. At this pressure, the total heat in a pound of steam is $318.2 + 872.3 = 1190.5$ B.t.u. The heat of the liquid at the atmospheric pressure is

179.3, therefore the heat to be charged to the engine per pound of dry steam used is $1190.5-179.3=1011.2$ B.t.u. The dry steam used per hour per i. hp. is $2970/97.9=30.35$ pounds. The work delivered per hour per horsepower $=33000\times 60$ foot-pounds. The foot-pounds of energy equivalent to the B.t.u. supplied per hour per horsepower is $778\times 1011.2\times 30.35$. Therefore, since the efficiency is the out-put divided by the in-put,

$$\textit{thermal efficiency} = \frac{33000\times 60}{778\times 1011.2\times 30.35} = .0828 = 8.28\%.$$

This is based on the i. hp. The dry steam used per hour per b. hp. is $2970/89.7=33.1$ pounds; hence the thermal efficiency based on the b. hp. is

$$\frac{33000\times 60}{778\times 1011.2\times 33.1} = 7.6\ldots$$

In this solution the factor $33000\times 60/778=2545$ will always occur in the equation for thermal efficiency. Hence the formula may be written in the form

$$\textit{thermal efficiency} = \frac{2545}{n\times \textit{B.t.u. per pound of dry steam}},$$

where n is the number of pounds of dry steam used per hour per horsepower.

The thermal efficiency of a steam engine will seldom, if ever, exceed 25 per cent. This may seem to be a very low value, but it is impossible for the engine to use a very large part of the heat supplied due to the fact that the exhaust steam carries with it its heat of vaporization. On account of condensation of steam in the cylinder and other causes of heat loss, the efficiency of a reciprocating engine seldom approaches the efficiency of an ideally perfect engine working under the same range of pressure, but a well-designed steam turbine of large size may do so.

95. Cylinder Condensation. — The largest single loss in the average engine is due to what is known as ***initial condensation.*** Since the cylinder walls are made of iron, which is a good conductor of heat, they naturally absorb heat from any hotter body or substance placed in contact with them and they give up heat to a cooler body. The steam comes into the cylinder at a relatively high pressure and temperature. Both the pressure and the temperature drop in the cylinder, and the steam leaves at a relatively low pressure and temperature. Since the cylinder walls are exposed first to hot, and then to cool steam, their temperature will never be as great as that of the incoming steam, nor as low, during operation, as that of the outgoing steam.

When the steam first enters the cylinder and strikes the cooler walls, a part of its heat will be absorbed by the walls. This

causes a partial condensation of the steam. Since the engine operates by virtue of the steam pressure and volume, it is readily seen that a shrinkage in volume causes a loss of work, and a lowering of efficiency. By the time the steam leaves the cylinder, it is cooler than the cylinder, and it takes back some of the heat it gave to the walls, but at too late a time to avoid the loss in efficiency. Depending upon the type of engine and the conditions of operation, the condensation may continue until release occurs, or re-evaporation may start during the expansion of the steam between cut-off and release. By computing from the indicator diagram the weights of steam at cut-off and at release, we find a net condensation during expansion if the weight at release is less than at cut-off, and a net re-evaporation if the weight at release is greater than at cut-off. The computation of condensation or re-evaporation during expansion is of little value since most of the re-evaporation occurs after release and before the steam leaves the exhaust ports.

96. Steam Accounted for by the Indicator Diagram. — The A. S. M. E. code for testing steam engines calls for the computation of the steam accounted for by the indicator diagram at points near the cut-off and release. Mark the points of cut-off and release and a point on the compression curve where we are sure the exhaust valve is closed, as in Fig. 47. Find the ratio of stroke at these points. The volume back of the piston at cut-off is the ratio of stroke at cut-off plus the ratio of clearance, shown by a, times the piston displacement. Scaling the pressure at cut-off from the diagram, we may compute the weight of dry steam back of the piston at cut-off by means of steam tables.

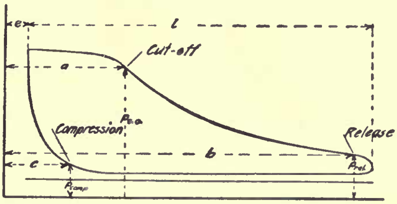


FIG. 47

Not all of the steam back of the piston at cut-off entered on that one stroke from admission to cut-off, since some of it was in the cylinder during compression. The amount that was admitted is the weight at cut-off minus the weight caught at compression. The weight of steam compressed may be computed in a manner similar to that at cut-off. The weight of steam per hour accounted for by the indicator diagram is then equal to

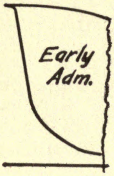


FIG. 48

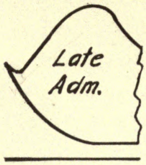


FIG. 49

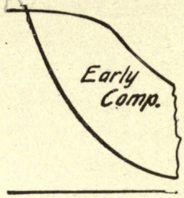


FIG. 50

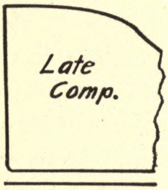


FIG. 51

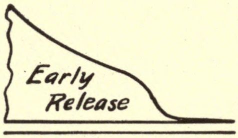


FIG. 52

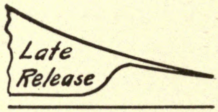


FIG. 53

Unequal Cut-Off

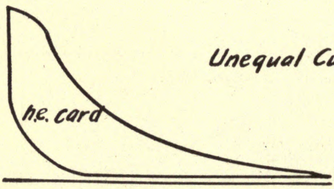


FIG. 54

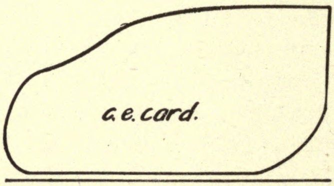


FIG. 55

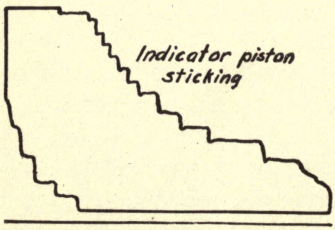


FIG. 56

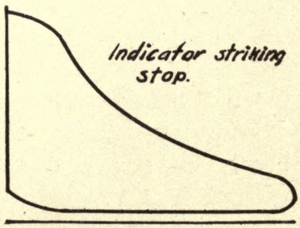


FIG. 57

$(W_{c.o.} - W_{comp.}) \times N \times 60/\text{i. hp.}$, where $W_{c.o.}$ and $W_{comp.}$ are the weights of steam back of the piston at cut-off and compression, respectively. The weight is calculated separately for the head-end and crank-end of the cylinder, and the two values added to give the total for the engine. The weight accounted for at release is computed in the same way, but the two results usually differ slightly on account of the net condensation or the re-evaporation during expansion from cut-off to release. The weight of steam accounted for by the indicator diagram will be considerably less than the actual amount used by the engine, because of the initial condensation of steam when it first enters the cylinder.

97. Valve-setting from the Indicator Diagram. — It has been mentioned previously that one of the uses of the indicator is to assist in the setting of the valves While the subject of valve-setting will not be discussed thoroughly here, faulty setting may be recognized from the appearance of the diagram. Figure 48 shows the effect when the admission valve opens too soon.

Figure 49 shows the results of late admission. Due to the tardiness of the valve opening, the steam is throttled, and the pressure for a large part of the stroke is lowered considerably.

Figure 50 shows the effect of early compression. The steam caught when the exhaust valve closes is compressed to a pressure above that in the steam chest, the admission valve is lifted off its seat, and some of the steam escapes into the steam chest.

Figure 51 shows almost no compression. This would cause no harm in a very slow-speed engine, but with higher speeds the steam caught at compression acts as a cushion and makes for smooth running.

Figure 52 shows too early a release, and Fig. 53, too late a release. Both cause a loss in the area of the diagram.

Figures 54 and 55 show unequal cut-off in the two ends of the cylinder. The crank-end is doing a much larger proportion of the work. The work done by the two ends should be about equal.

Figure 56 shows improper lubrication of the indicator piston or the binding of some part. A wavelike motion of the curve is sometimes noticed when the diagram is taken from a high-speed engine, due to the vibration of the indicator spring, but it differs materially from Fig. 56. In Fig. 57, the indicator drum is striking the stop on account of improper adjustment of the length of the cord connecting the indicator and the reducing mechanism.

CHAPTER VII

COMMON TYPES OF STEAM ENGINES

98. Slide-valve Engine. — Where simplicity and reliability are of more importance than high efficiency, the slide-valve engine is used. The simplest type of slide-valve was shown in Fig. 39, and the principles of its operation were explained to some extent in the previous chapter. There are many varieties of slide-valves, the more common of which will be described later. Most of the smaller stationary engines in use are equipped with the slide-valve, and all American locomotive engines are of this type.

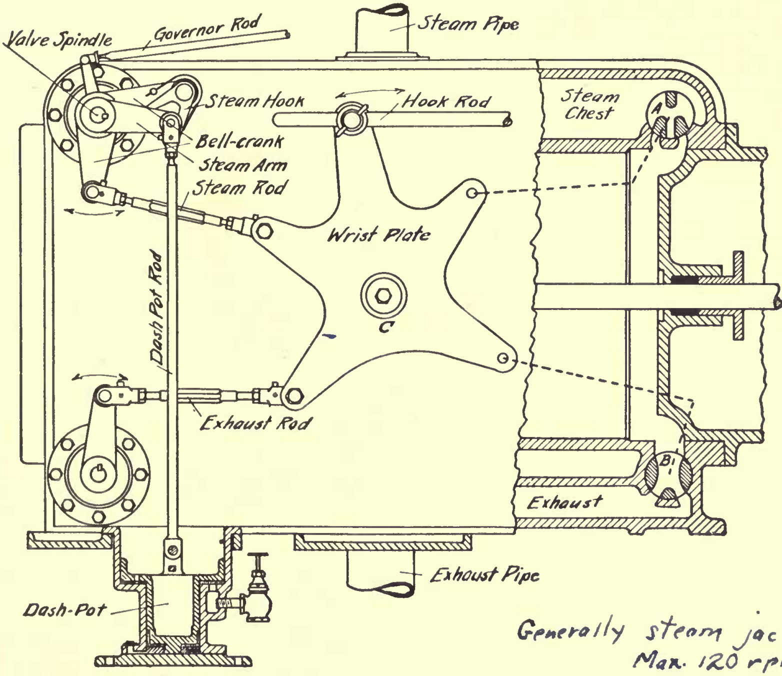
While the plain slide-valve is very simple, it has certain defects. One of these is the impossibility of obtaining the proper steam distribution at all loads, *i.e.* of making the events of stroke occur at the proper place to give the highest efficiency at light load and also at heavy load. Various modifications and improvements have been made on the slide-valve to remedy this defect, the chief one of which is to place a second slide-valve on top of the main valve, and to control cut-off by a rider.

Another defect of the plain slide-valve is the slowness with which the steam ports open up and close at some loads, which cause what is known as *wire-drawing*. This is simply a throttling of the steam by the valve as it enters the cylinder. This throttling usually causes a lowering of the efficiency of the engine.

99. The Corliss Engine. — By far the most common type of high-grade reciprocating stationary steam engine in this country is the Corliss engine. The name comes from its inventor and first producer, George Corliss, an engineer and engine builder of Providence, R. I. There are two distinguishing features of this engine. The first of these is the oscillating cylindrical valve. The second is the means for disengaging the valve from the mechanism that drives it, and the quick closing of the valve after its disengagement. To understand the Corliss valve mechanism thoroughly, it is necessary to make a rather thorough analysis of its motion. We shall not do this in this chapter. The general principle of operation of the gear is fairly simple, however.

Figure 58 shows a typical Corliss engine cylinder. The right end is cut in section to show the construction of the valves and

their locations relative to the cylinder. The left end shows an ordinary form of the mechanism that moves the steam valves and the exhaust valves. An eccentric on the shaft is connected to the hook rod which operates the valves through an eccentric rod and rocker arm. This gives the hook rod a horizontal reciprocating motion that is nearly harmonic. The hook rod is attached



Generally steam jacketed
Max. 120 rpm.

FIG. 58

to a wrist plate which is pivoted to the cylinder at C. The wrist plate is thereby given an oscillating motion.

Four rods are attached to the wrist plate. The two upper rods are the steam rods, which transmit the motion to the two steam valves, and the lower or exhaust rods drive the two exhaust valves. The steam rods are attached to bell cranks or double arms that are pivoted on the valve spindle but are not attached to it. It is thus possible for the wrist plate and the bell crank

to move without affecting the valve in any way. The cylindrical steam valve has a spindle which extends out of the steam chest. To the outer end of this spindle there is keyed a steam arm. Any motion of this arm causes the valve to move. A block is attached to the back side of the steam arm, and a hook which is carried by the upper arm of the bell crank catches over it. As the steam rod moves to the right, the steam arm is picked up and the valve is turned. After having lifted the steam arm a certain distance, the hook is made to disengage with the block and the steam arm is released.

The steam arm is connected to the piston of a dash-pot by a *dash-rod.* As the steam arm is raised, a partial vacuum is formed in the dash-pot. When the steam arm is released from the hook, it is suddenly pulled downward by the vacuum in the dash-pot. As the steam arm is lifted, the valve opens and admits steam to the cylinder. When it is pulled down, the valve is closed suddenly, giving a quick cut-off. The time at which the hook is made to release the steam arm is controlled by a cam whose position is regulated by the governor. This cam engages with the tail of the hook and causes the disengagement. At light loads, the trip occurs soon and an early cut-off is given, and the cut-off is retarded as the load of the engine is increased.

Figure 59 shows the trip mechanism on a larger scale. *DAC* is the bell crank. As the point *D* moves backward and forward in a nearly horizontal direction, the point *C* moves up and down in a nearly vertical direction. The pin *C* carries the steam hook. The tail of the hook engages with the knock-off cam, and its jaw engages with the block attached to the steam arm at *B*. For any one load the knock-off cam is stationary, and as *C* goes up, the tail of the hook is pushed away from *A* by the cam, which causes the latch to disengage with the block *B*. When there is a heavy load on the engine, the governor rod moves to the left, which raises the knock-off cam and makes the trip come later, giving a long cut-off. There is also a safety cam, shown in Fig. 59. If the governor fails to rotate, the safety cam comes into contact with the tail of the hook and prevents the picking up of the steam arm and therefore causes a failure to admit steam to the cylinder.

There is no disengagement between the exhaust arm and the exhaust valve, so that the events of release and compression occur

at the same ratio of the stroke for all loads. Both the steam valve and the exhaust valve of Fig. 58 are double-ported, which gives twice the opening for the passage of steam with the same valve movement as with the single-ported type.

From the description just given it is seen that cut-off is inde-

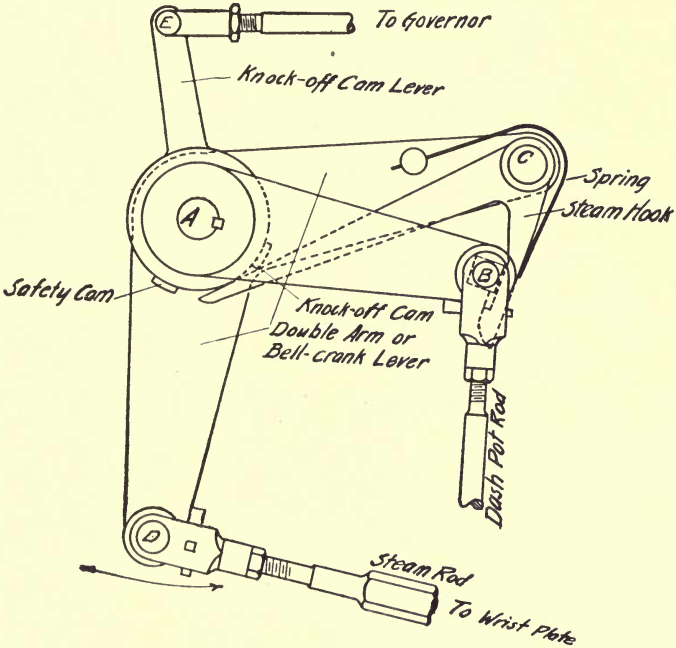
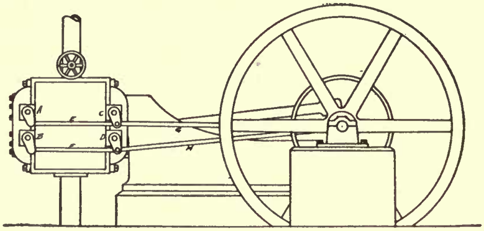


FIG. 59

pendent of the other events, and that the steam valve closes quickly, thereby preventing wire-drawing at closing. If a careful analysis of the motion is made, it will be seen that the steam valve opens the port nearly as widely at light loads as at full loads. The force necessary to operate the valve mechanism is not large and the work done in moving the valves is a very small part of the total output of the engine.

100. The Four-valve Engine. — With a single slide-valve, the changing of one event necessitates the changing of all the others. To avoid this difficulty, engines are often made that

have four valves, a steam valve for each end of the cylinder, and an exhaust valve on each end. The exhaust valves are driven by a fixed eccentric, so that the release and compression are the same at all loads. The steam valves are controlled by the governor; hence cut-off and admission will vary for different loads.



f FIG. 60

Figures 60 and 61 show one style of four-valve engine. This particular engine has oscillating cylindrical valves similar to those shown in Fig. 58 for the Corliss engine. Some makers, probably to make use of the enviable reputation of the Corliss engine, call this type a non-releasing Corliss. This engine lacks, however, the distinct advantage of the trip found in the true Corliss type.

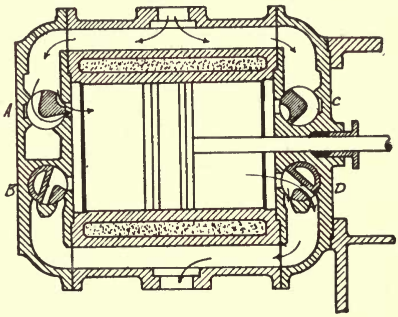


FIG. 61

101. The Compound Engine. — When all the expansion of steam takes place in one cylinder, we have what is known as a simple engine. If the steam passes through two successive cylinders, the engine is said to be a ***compound engine.*** If there are three successive cylinders, it is called a ***triple-expansion engine.***

If there are four successive cylinders, it is called a ***quadruple-expansion engine,*** etc. An engine may have two cylinders and not be compound, *i.e.* it may be a twin-cylinder engine, in which half the steam passes through one cylinder and half through the other. Likewise a compound engine may have three cylinders, all the steam passing through one high-pressure cylinder, and then dividing, half passing through each of the two low-pressure cylinders.

The purpose of compounding is to reduce the initial condensation. It does not necessarily follow that there is a greater ratio of expansion of steam in a compound engine than in a simple engine, since that depends upon the point of cut-off. We have seen that initial condensation is caused by the range in temperature within the cylinder. The temperature range is less in each cylinder of a compound engine than in the single cylinder of a simple engine of the same capacity. Since the amount of condensation does not vary directly as the total temperature range, there may be considerably less total condensation if the steam is passed through two successive cylinders than if all the expansion occurred in one cylinder.

Several years ago the idea of compounding was very popular and was carried to the extreme. Many triple-expansion engines, and some quadruple-expansion engines, were built. Experience proved, however, that there was a practical limit to which the idea might be carried. Now stationary engines are seldom built with more than two pressure-stages, except in direct-acting pumps. In the marine service, the triple-expansion engine is still popular, partly for the reason that it is desirable to have three cranks on the same shaft to give a greater uniformity of torque on the propeller shaft, and partly on account of the uniformity of load on marine engines. Of the many types of compound engines that have been built, only two are in common use in land service at present. We shall proceed to consider these.

102. The Tandem-compound Engine. — In the tandem-compound engine, the pistons of the two cylinders are placed on the same piston rod, as shown in Fig. 62. The cylinder to the left is the high-pressure cylinder, and the one to the right is the low-pressure cylinder. The steam ports of the high-pressure cylinder are at *a* and *b*; the exhaust ports of the low-pressure cylinder are

at *c* and *d*. The pistons, in Fig. 62, are shown moving to the right. Steam is entering the high-pressure cylinder through *a*, and leaving it through *f*. The exhaust steam from the crank end of the high-pressure cylinder passes to the head end of the low-pressure cylinder either directly or through a stationary vessel called a *receiver*, *i.e.* the back pressure on the piston *A* is the forward pressure on the piston *B*. On the return stroke, steam

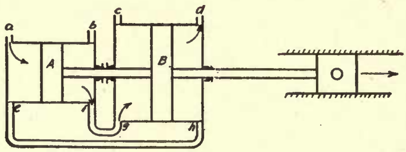


FIG. 62

enters the port *b* and the exhaust from the high-pressure cylinder leaves through *e* and enters the low-pressure cylinder at *h* either directly or through the receiver. With the tandem arrangement only one cross-head, connecting rod, crank, and frame are needed.

In locomotive work, the Baldwin or Vauclain compound engine is sometimes seen. In this engine, the cylinders are placed side by side, and both piston rods attach to the same cross-head. The method of steam distribution is similar to that of the tandem type. Figures 63 and 64 show the high-pressure and low-

FIG. 63

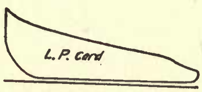


FIG. 64

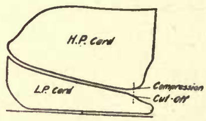


FIG. 65

pressure indicator diagrams, as taken from a Baldwin compound engine. The high-pressure card comes from the crank end of the high-pressure cylinder and the low-pressure card from the head end of the low-pressure cylinder. These cards were taken with springs of different scales. Figure 65 shows the same diagrams when drawn to the same scale of pressure. It is noticed that the back-pressure line of the high-pressure card parallels the admission line of the low-pressure card from the left end up to the point of cut-off in the low-pressure cylinder. The reason for this is

obvious, since the exhaust from the high-pressure cylinder passes directly to the low-pressure cylinder. When the admission valve of the low-pressure cylinder closes, compression must necessarily start in the high-pressure cylinder. Since it is necessary, with such a high pressure at exhaust as exists in the high-pressure cylinder, to have the compression occur late, it follows that cut-off must come very late in the low-pressure cylinder. This is not an ideal condition, but it is necessary if no receiver is placed between the two cylinders. If a receiver were placed between the two cylinders so that it could act as a reservoir into which to discharge, and from which to draw steam, it would not be necessary to have the preceding relation between compression and cut-off.

103. The Cross-compound Engine. — In the cross-compound engine (Fig. 66), each cylinder has its own cross-head, connect-

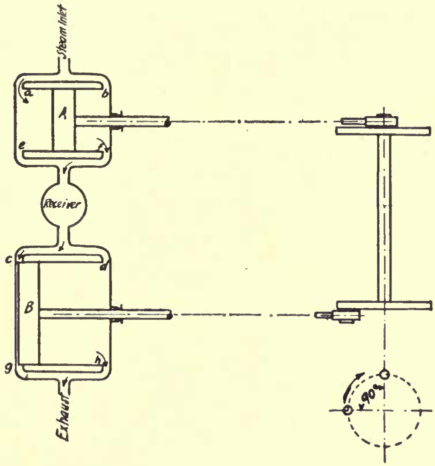


Fig. 66

ing rod, crank, and frame. The cranks are usually spaced 90° apart. A by-pass is arranged so that the engine can be started even if the high-pressure cylinder stops on dead center, by admitting steam directly to the low-pressure cylinder.

Each cylinder has its own valve mechanism, and the exhaust from the high-pressure cylinder passes into a receiver from which the low-pressure cylinder takes its steam. This arrangement permits a better steam distribution than that used in the tandem

type without a receiver. If the receiver is quite large, the back pressure in the high-pressure cylinder during exhaust will be nearly constant. Figures 67 and 68 show the indicator diagrams from a cross-compound engine. It should be noticed that the engine exhausts into a condenser.

104. Cylinder Ratio. — The cylinder ratio of a compound engine is the ratio between the piston displacements of the low-pressure and the high-pressure cylinders. While it is not essential that the length of stroke be the same for both high-pressure and low-pressure cylinders of a cross-compound engine, they are made so. The cylinder ratio is then the ratio of the squares of the diameters of the low-pressure and high-pressure cylinders.

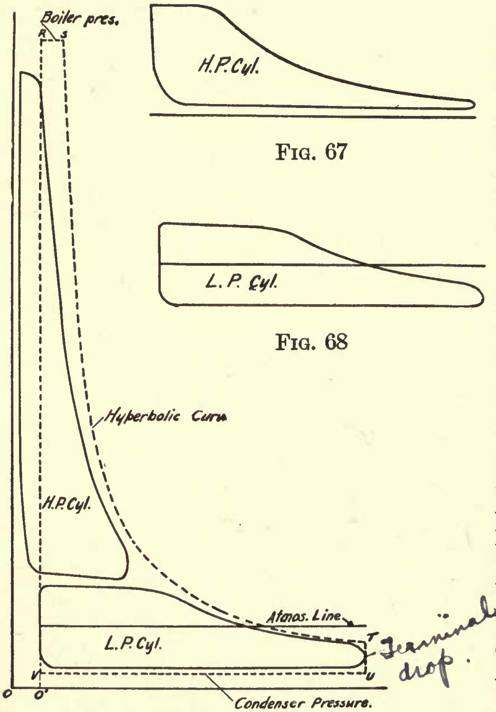


FIG. 67

FIG. 68

FIG. 69

105. The Combined Indicator Diagram. — The combined diagram is constructed by plotting both cards to the same scale of pressure and volume. Usually we do not change the low-pressure diagram but change the scale of the high-pressure card to conform to it. Figure 69 shows the combination of diagrams of Figs. 67 and 68. The low-pressure diagram is identical with Fig. 68, while the length of the high-pressure diagram equals the length of the low-pressure diagram divided by the cylinder ratio. The high-pressure diagram is placed to the right of the pressure axis its clearance distance, *i.e.* its distance from the axis equals the ratio of the high-pressure clearance times the new length of the high-pressure diagram.

106. Diagram Factor.— The definition of the ***diagram factor*** as given in the 1915 edition of the A. S. M. E. Power Test Code is as follows:

> The diagram factor is the proportion borne by the mean effective pressure measured from the actual diagram to that of a hypothetical diagram which represents the maximum power obtainable from the steam accounted for by the actual diagram at the point of cut-off; assuming first, that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during admission or release; third, that the expansion line is a hyperbolic curve; and, fourth, that the initial pressure is that of the boiler, and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

To determine the steam accounted for by the actual diagram at the point of cut-off, draw hyperbolic curves through the point of compression P and the point of cut-off O (Fig. 70) until they

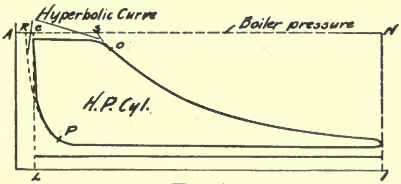


FIG. 70

cut the boiler-pressure line at R and S. The length of RS is the length of the admission line for the hypothetical diagram, FA in Fig. 42, drawn to proper scale. The hypothetical diagram is drawn as in Fig. 42 except that the boiler pressure is taken as the initial pressure, release comes at the end of the stroke, the back pressure is the atmospheric pressure (condensing pressure in a condensing engine), there is no compression, and there is no clearance. The hypothetical diagram for the combined diagrams of Fig. 69 is shown dotted. Since we assume there is no clearance, the length of the hypothetical diagram is equal to that of the low-pressure card. The distance RS at boiler pressure is determined from the high-pressure diagram, as in Fig. 70. From S to T construct a hyperbolic curve, using the origin O', and not O. Release is at the end of the stroke and the back-pressure line is at the condenser pressure.

In Fig. 69, the mean effective pressure of the combined diagrams and of the hypothetical diagram are in the same ratio as the areas of the combined and hypothetical diagrams, because they are of the same length. To find the diagram factor of the combined cards, divide their area by the area of the hypothetical diagram.

107. Ratio of Expansion. — The A. S. M. E. Power Test Code gives the following rule:

> To find the percentage of cut-off, or what may best be termed the *commercial cut-off,* the following rule should be observed:
>
> Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete, draw a hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured up to this point by the total length.
>
> l + c / i + c
>
> l at release
> clearance
> l at cutoff
>
> To find the *ratio of expansion,* divide the volume corresponding to the piston displacement (Release), including clearance, by the volume of the steam at the commercial cut-off, including clearance.
>
> In a multiple-expansion engine the ratio of expansion is found by dividing the volume of a low-pressure (Release) cylinder, including clearance, by the volume of the high-pressure cylinder at the commercial cut-off, including clearance.

108. The Unaflow Engine. — The unaflow, or uniflow, engine is shown diagrammatically in Fig. 71. There is an admission valve at each end of the cylinder. The exhaust steam escapes through a port located around the circumference of the cylinder midway between the two ends. The piston, which is longer than in most engines, itself uncovers the exhaust port at about 90 per cent of the stroke. Compression must start when the piston is at the same place on the return stroke. Under non-condensing conditions this would give a very excessive compression pressure; hence the engine normally is run condensing, under which conditions the compression pressure is moderate. The thermal efficiency is about the same as that of a compound engine. The gain in efficiency over the ordinary double flow engine is due to the reduction of initial condensation. The condensation

is reduced with the unaflow principle because the ends of the cylinder are kept hotter than the central portion. High-pressure steam never comes in contact with the central part of the cylinder and the flow of steam is from the ends toward the middle. The exhaust steam passing out through the central port does not cool the walls as much as it would if it flowed back to

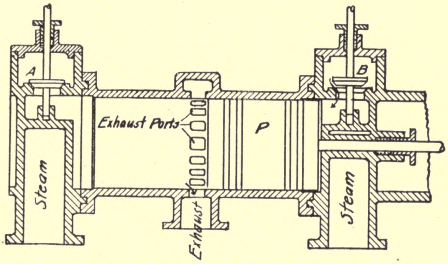


FIG. 71

the ends of the cylinder upon leaving. In actual engines, provision must be made for relieving the excessive compression pressure, should the vacuum break. This is done by a relief valve that adds to the clearance or allows the compressed steam to re-enter the steam chest, or by adding an auxiliary exhaust port nearer the end of the cylinder, which is opened automatically when the vacuum fails.

CHAPTER VIII

VALVES

109. Introduction. — From our previous study of the steam engine we have learned that the purpose of the valve is to admit steam to the cylinder, and to release steam from it. The time at which the events occur must be such that the engine is capable of doing the work required, and that it may have as high an efficiency as possible under the conditions of operation. An engine may run with the valves improperly set or designed, but more steam will be used than if the valves functioned properly.

110. The *D* Slide-valve. — The engine of Fig. 39 has what is commonly known as a *D* slide-valve. The valve slides back and forth on its seat, alternately opening and closing the ports. An eccentric on the shaft drives the valve. The eccentric rod is usually quite long in comparison with the throw of the eccentric, so that the valve may be considered to have the same motion as the horizontal component of the eccentric.

In Fig. 72*a*, the valve is shown in *mid-position;* consequently the eccentric will either be directly above or directly below the center of the shaft. In mid-position, the valve laps over the edges of the port; the amount it extends over on the steam side is called the ***steam lap,*** and on the exhaust side, the ***exhaust lap.***

At the right side of Fig. 72*a* is shown the relative position of the eccentric and the crank. The eccentric leads the crank by angle θ. This angle θ will be the same at all times during the revolution. In Fig. 72*b*, the crank is on head-end dead center, and the valve is uncovering the head-end port a small amount. The amount the port is open when the crank is on dead center is called the ***lead.*** It is measured in inches. The valve in Fig. 72*b* is to the right of its mid-position by an amount equal to the *steam lap plus the lead.* In moving from the position shown in Fig. 72*a* to that in Fig. 72*b*, the eccentric has moved a horizontal distance equal to the *steam lap plus the lead*, and has turned through a certain angle which is called the ***angle of advance.*** It is evident that the eccentric is ahead of the crank by an angle of 90° plus the angle of advance. It is customary to speak of the angle of advance and not of the whole angle θ.

Figure 72*c* shows the valve at head-end admission. The valve

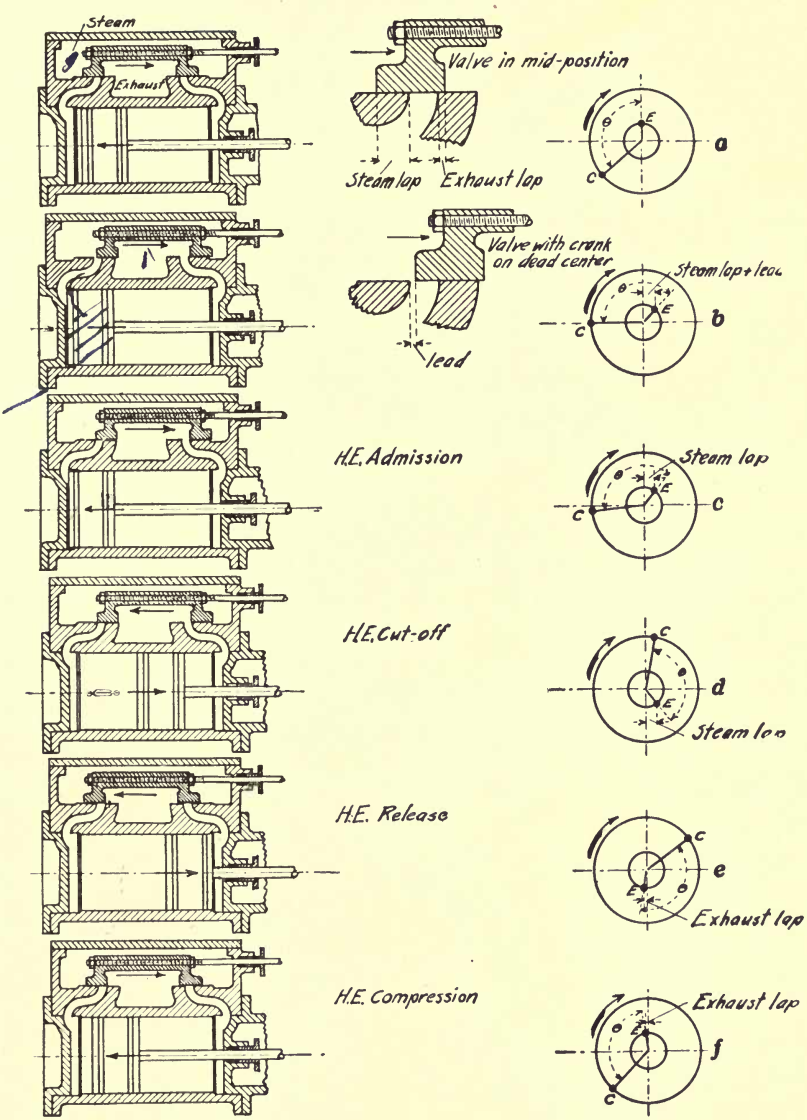

Steam
Exhaust
Valve in mid-position
Steam lap
Exhaust lap
a
Valve with crank on dead center
Steam lap + lead
lead
b
H.E. Admission
Steam lap
c
H.E. Cut-off
d
Steam lap
H.E. Release
e
Exhaust lap
H.E. Compression
Exhaust lap
f

FIG. 72

is on the point of opening the head-end port for the admission of steam, and is traveling to the right. In the admission position it is to the right of its mid-position by a distance equal to the steam lap.

Likewise the eccentric will be to the right of its mid-position by a horizontal distance equal to the steam lap. The crank is back of the eccentric by the angle θ and is seen to be approaching its head-end dead-center position. If there is any lead, the crank will never be quite up to its dead-center position at admission.

At head-end cut-off, Fig. 72*d*, the valve is to the right of its mid-position by a distance equal to the steam lap and its direction of motion is to the left. The position is the same as for admission, but it is going in the opposite direction. The eccentric is now below the center line of the shaft.

Figures 72*e* and 72*f* show the relative positions at head-end release and compression. At both of these events, the valve is at the left of mid-position by an amount equal to the exhaust-lap distance. It is moving to the left at release and to the right at compression.

In all the six diagrams of Fig. 72, the positions of the piston and its direction of motion are shown. The cylinder section in each diagram is in a horizontal plane and therefore is at an angle of 90° from the diagram showing crank and eccentric positions

For an understanding of the slide-valve and the analyses to follow, it is essential that the student have a precise conception of the relative positions of the valve on the seat, the eccentric and the crank relative to the center positions, and the position of the piston in the cylinder.

111. Relative Motion of Crank and Piston. — Since the connecting rod is not very long compared with the crank arm, we cannot assume that the horizontal movement of the crank is the same as the piston movement. This is clearly seen from the diagram in Fig. 73. As the crank moves from A to C, the cross-head moves from E to D. That part of the stroke completed by the cross-head is a'. It is evident that a' is considerably larger than the horizontal movement of the crank in going from A to C.

In our analysis of valve motions, it is not customary to draw

in the cross-head D to find the proportion of the stroke at different crank positions, but the following scheme is used. The horizontal diameter of the crank circle AB is extended to the left. With the length of connecting rod DC as a radius at the desired scale, and with D as a center, strike the arc shown by the dotted line CG. This gives the distance $AG = a$, on the diameter of the crank circle, that is equal to $ED = a'$, the movement of

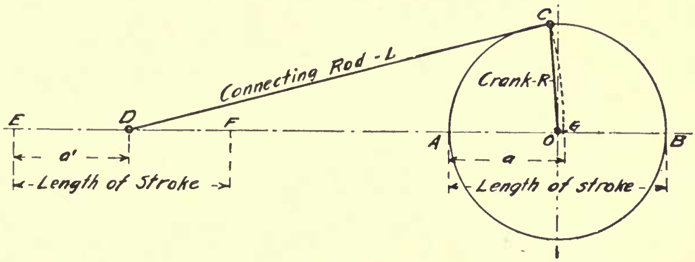


FIG. 73

the piston or cross-head from E to D. In the valve analysis it is not necessary to draw the crank circle to any particular scale if we keep the proper ratio between the lengths of the crank and the connecting rod. This ratio usually is expressed as R/L. No matter what the scale of the crank circle may be, the ratio of a to the length of stroke will be constant.

112. Valve Diagrams. — Many diagrams have been used to show graphically the relation of the movement of the valve to the movement of the piston, or of the relative movements of eccentric and crank. Only those most commonly used in this country will be explained here, *i.e.* the ***valve ellipse,*** the ***Bilgram diagram,*** and the ***Zeuner diagram.***

113. The Valve Ellipse.—In this diagram the system of rectangular coordinates is used. The valve displacement is plotted vertically and the piston displacements are plotted horizontally. On the left of Fig. 74 is shown a crank and eccentric. The eccentric is ahead of the crank by 90° plus α.

With the crank at C, the piston is at a distance x from the center of the stroke. At the same time, the valve is at a distance y from its mid-position. If we plot x against y, we get a point G.

The coordinate axes are the horizontal and vertical diameters of the crank circle.

On the right of Fig. 74 this same operation is carried out for twelve crank positions with their corresponding eccentric positions. The crank positions are denoted by C_1, C_2, C_3, etc., and the corresponding eccentric positions by E_1, E_2, E_3, etc. Plotting the displacements, we get the points 1, 2, 3, etc. Connecting the points thus found by a smooth curve, we get what is known as a ***valve ellipse.*** It is evident from Fig. 74 that this is not a true

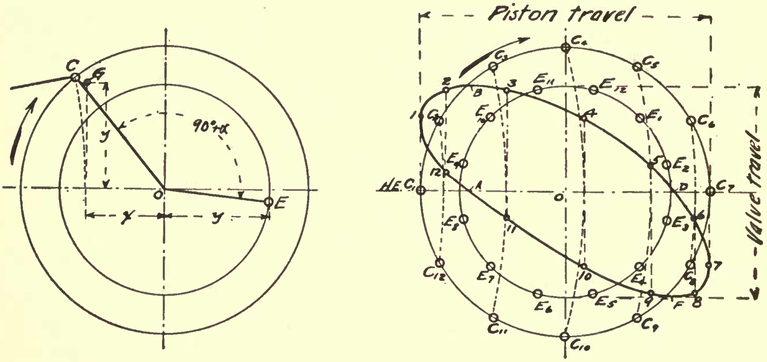


Fig. 74

ellipse. It would have been but for the distortion due to the short length of the connecting rod.

The upper half of the ellipse ABD represents the valve movement to the right of its mid-position. The lower half, DFA, represents the valve movement to the left of its mid-position. From A to B it gives the movement from mid-position to extreme right; from B to D, from the extreme right to mid-position; from D to F from mid-position to extreme left; and F to A, from the extreme left to mid-position.

The valve ellipse of Fig. 74 is reproduced in Fig. 75. Four horizontal lines are drawn through the ellipse. The head-end steam lap is the distance from the top line to the horizontal axis. The crank-end steam lap is the distance from the bottom line to the axis. The head-end exhaust-lap line is drawn below the axis and the crank-end exhaust-lap line above. When the valve has moved to the right a distance equal to the head-end steam lap, head-end admission takes place. Admission is shown by the

point H on the ellipse, and the crank position corresponding to H is determined by projecting vertically from H to the axis.

As the valve moves from its extreme right position back to mid-position, head-end cut-off takes place. This is shown by the point I on the ellipse. The crank position corresponding to I is found by projecting down from I to the axis and striking an arc upward from this point to the crank circle. The radius of the arc is the length of the connecting rod. In like manner, the

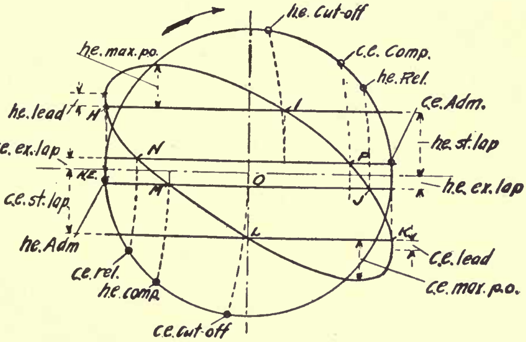


FIG. 75

crank position at head-end release is found from the intersection of the head-end exhaust-lap line with the ellipse at J.

The head-end compression point is at M, where the exhaust-lap line cuts the ellipse. The crank-end events are determined from the points K, L, N, and P. The vertical distance from the top point of the ellipse to the head-end steam-lap line is the head-end maximum port-opening, and the head-end lead is given by the distance from the extreme left point of the ellipse to the head-end steam-lap line. The crank-end maximum port-opening and lead are found in a similar manner.

In actual use, the ellipse is rather burdensome because it takes considerable time to construct it. It is evident also that the crank position at admission is not easily determined with accuracy. The ellipse is little used except in locomotive work.

114. The Bilgram Diagram. — On the left of Fig. 76, the crank and the eccentric are shown by C and E, respectively. The displacement of the valve from mid-position is y. The distance y is laid off perpendicular to the crank, and a line is drawn parallel to the crank at a distance y from it.

At the right of Fig. 76, this has been done for twelve crank positions. It is seen that these lines all pass through two points P and P', and that a line drawn from P to P' passes through the center of the crank circle and makes an angle α with the horizontal. Hence the perpendicular distance from the points P or P' to the crank at any position is the distance that the valve is from mid-position. The points P and P' are called the construction points in the Bilgram diagram.

Figure 77 shows the application of the Bilgram diagram. About the point P draw two circles, one whose radius is equal to the head-end steam lap, and the other with a radius equal

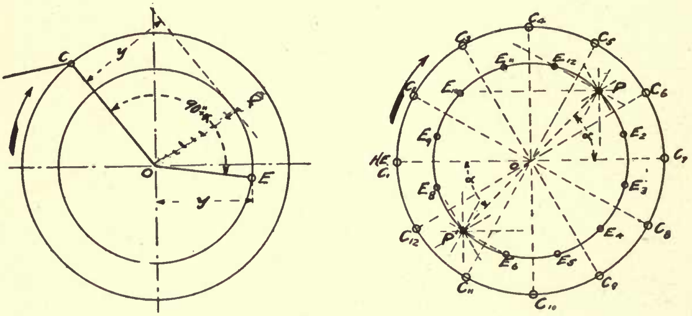


Fig. 76

to the head-end exhaust lap. Draw the crank-end lap circles with center at P'. The crank positions tangent to these lap circles give the positions at the different events. Head-end admission occurs when the valve is at a distance from its mid-position equal to the head-end steam lap, and when the valve is going away from its mid-position. The head-end admission position shown in Fig. 77 fulfills these conditions. At that time the crank is at a distance from the point P equal to the head-end steam lap, and further motion moves it farther from P. The crank position at head-end cut-off is tangent to the head-end steam-lap circle on the other side, *i.e.*, the valve is then at a distance equal to the steam lap from mid-position and further motion brings the valve nearer mid-position.

The crank positions for the other events are shown in Fig. 77. Reasoning similar to the preceding will show them to be correct. It is customary to draw the head-end lap circles about P, and the

crank-end lap circles about P', although there is no inherent reason for so doing.

Half of the valve-travel minus the steam lap equals the maximum port opening if the valve has no *over-travel, i.e.* if it does not move beyond the far edge of the port. Therefore the maximum port-opening is as shown in the figure. It is remembered that the port is open a distance equal to the lead when the crank is on dead center. In other words the valve is then at a distance equal to the steam lap plus the lead from its mid-position. Therefore the perpendicular distance of P from the horizontal

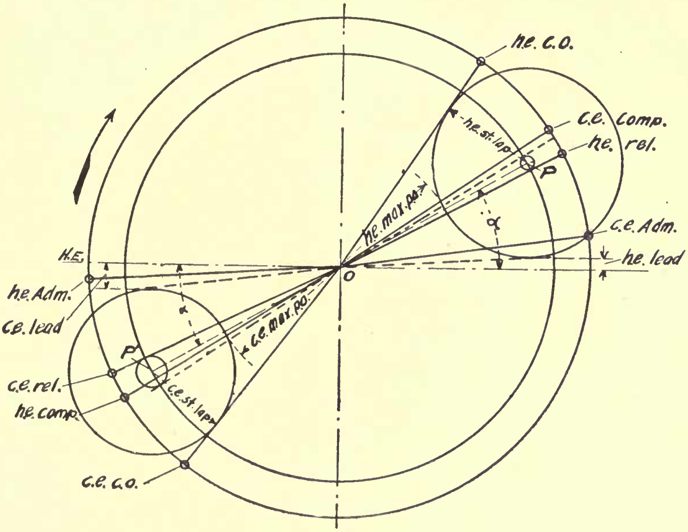


FIG. 77

axis is equal to the steam lap plus the lead. The distance of the steam-lap circle from the axis is then the lead.

115. The Zeuner Diagram. — On the left of Fig. 78, the valve displacement y is laid off radially on the crank from the center outward. This gives a point G on the crank. This has been done for twelve crank positions on the right of Fig. 78 and the points connected by a smooth curve. The points fall on the circumferences of two equal circles, the diameter of each of which is one-half the valve-travel. The line which forms the diameters of these two circles makes an angle α with the vertical. The

circle whose center is at A shows the movement of the valve to the right of its mid-position, and is called the ***right valve-circle.*** The other circle is called the ***left valve-circle***; it shows the movement of the valve to the left of the mid-position. When the crank is drawn in any position, the displacement of the valve is given by the distance from the center of the crank circle to the intersection of the crank with the valve circle.

The application of this diagram is shown in Fig. 79. With the crank on head-end dead center, the eccentric is at E, at an angle α to the right of the vertical. The diameters of valve circles, $P-P'$, are at an angle α on the other side of the vertical.

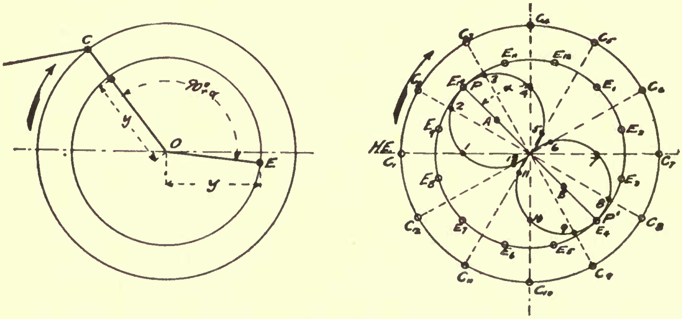


FIG. 78

The extremity P of the diameter of the valve circle is called the ***construction point*** in the Zeuner diagram. The lap circles are drawn as shown. The head-end steam-lap circle intersects the right valve-circle at the point T. The crank position through T is then head-end admission, because with the crank in this position the valve is at a distance equal to the steam lap to the right of mid-position. The crank position at cut-off is drawn through the point K, where the steam-lap circle intersects the right valve-circle. Head-end release and compression occur when the head-end exhaust-lap circle intersects the left valve-circle.

It may be proved by means of the similar triangles OWB and PKO or by actual construction that a line drawn from A to B is tangent to the steam-lap circle at W. This line is perpendicular to the diameter of the valve circles. In like manner a line drawn from I to F is tangent to the head-end exhaust-lap circle, and is perpendicular to the diameter of the valve circles. The same thing is true of the lines HG and JD for the crank end. It is

often better to draw the lines AB, IF, HG, and JD than to determine the crank positions at the events by the intersections of the valve circle and the lap circle.

ON is equal to the steam lap plus the lead, because the valve circle cuts the crank on dead center at N. But ONP is a right triangle, since it is inscribed in a semicircle. If QS is drawn parallel to AW, the triangle OSQ is a right triangle, and it is similar and equal to the triangle, ONP. Therefore OS is equal

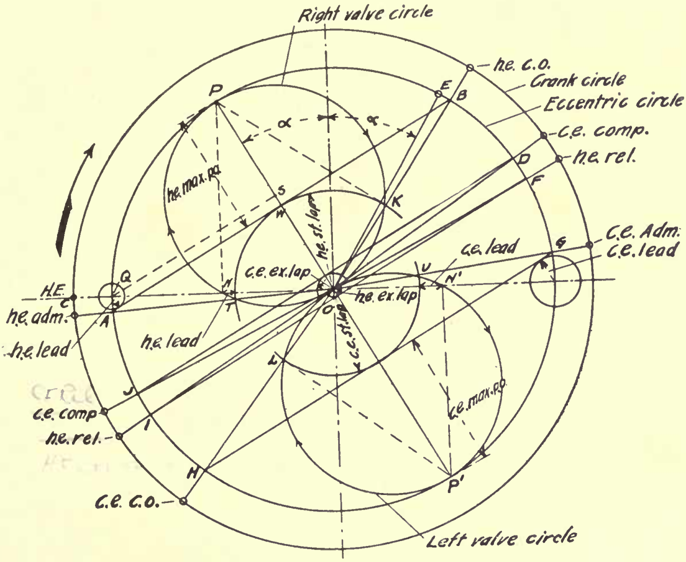


FIG. 79

to the steam lap plus the lead, and WS is equal to the lead. If we then draw a circle about Q as a center, tangent to the line AW, its radius is the lead. This is called the ***head-end lead circle.*** The ***crank-end lead circle*** is drawn about X tangent to HG. The head-end maximum port opening equals PW, which is one-half the valve-travel minus the steam-lap. The line PK is perpendicular to the crank at cut-off, because PKO is a right angle since it is inscribed in a semicircle.

The application of these valve diagrams to practical problems will show their value. Space will not be taken here to give the solutions of the various common problems in which these diagrams are used. The Bilgram and Zeuner diagrams are both

adapted to problems of valve setting, but the Bilgram diagram is the more convenient for use in designing valve gears.

116. Types of Slide-valves. — The simple *D* slide-valve has been discussed and its action explained. This type of valve is much used, but it has certain defects which have been overcome in other types. One of the defects of the simple *D* valve is the large force necessary to move it when high steam pressure is used. The steam pressure on the back of the valve presses it against the seat. This pressure times the coefficient of friction between the valve and the seat is the force that must be exerted to move the valve. The work done in operating the valve is the force times the distance the valve is moved. If either the force or the distance is decreased, the work necessary to operate the valve will be lessened.

117. Valve with Pressure Plate. — The pressure on the back of the valve may be removed by putting a pressure plate above it, somewhat in the manner shown in Fig. 80. A steam-tight fit between the valve and the plate is made by strips set into slots in the valve. These are pressed up against the plate by springs from beneath, and they act in much the same way as rings on a piston. If any steam leaks by the strips, it may escape to the exhaust through a vent in the valve.

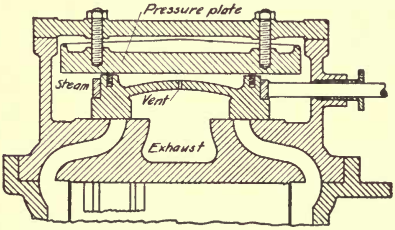


Fig. 80

This scheme enables us to remove as much of the pressure from the back of the valve as we desire, but some pressure downward is desirable in order to keep the valve firmly seated. Many flat slide-valves have pressure plates. Aside from removing the pressure, the plate does not affect the valve in any way.

118. The Piston Valve. — Instead of a flat valve such as we have considered, a ***piston valve*** is used extensively. Figure 81 shows a form of this type where the valve is cylindrical and slides in a cylindrical chamber. It is readily seen that it is perfectly balanced, since the steam causes no thrust either endwise or on the seat. Piston valves are very easy to operate, but are liable to leak steam when they become worn. Many of them

have rings similar to piston rings to keep this leakage of steam to a minimum. If rings are used it is necessary to bridge the ports. A broken ring is liable to cause severe damage and care must be exercised to keep them in good condition.

In Fig. 81, the steam is led to the inside of the valve, so that the steam lap is on the other side of the port from the valves previously considered. A valve so constructed is said to be an

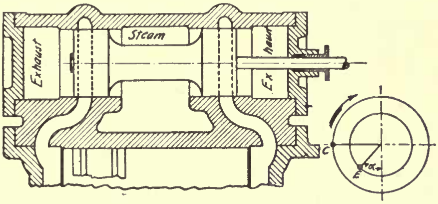


FIG. 81

indirect valve, or to have ***inside admission.*** Most piston valves have inside admission, but this is not a necessity. It is easier to keep the stuffing-box tight against exhaust steam than against high-pressure steam. With the inside admission arrangement, also, the live steam has less surface exposed to radiation. With an inside admission valve, the eccentric follows the crank by an angle of $90° - \alpha$. The valve diagrams studied will have the same form as before, but what was right-hand is now left-hand, *i.e.* in the Zeuner diagram, for instance, what was formerly the right valve-circle is now the left, but otherwise there is no change in the diagram and no difference is made in the solution of a problem.

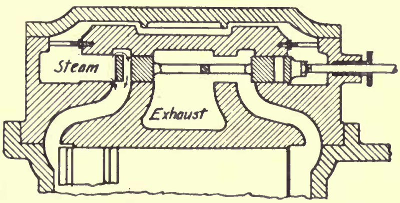


FIG. 82

Sweet Valve

119. Double-ported Valves. — As has just been mentioned, the work required to move the valve is the product of the force required to move it and the distance it is moved. The distance may be cut in half by making the valve double-ported. Figure 82 shows a so-called ***double-ported valve.*** It is seen that for a certain valve movement twice the area for the passage of steam is given in this type compared with a simple slide-valve. There are many different forms of double-ported valves, but they are much the same in principle as that shown in Fig. 82.

120. The Gridiron Valve. — If the idea of a double-ported valve is carried a step farther, we may get a very large aggregate opening for the passage of steam with but a small movement of the valve. Figure 83 shows a ***gridiron valve.*** In many valves of this type there are a large number of openings, whereas Fig. 83 shows only three. A valve of this character can have no exhaust functions, and separate exhaust valves must be provided. If one exhaust valve takes care of both ends of the cylinder, the engine is called a ***two-valve engine;*** if there is a separate exhaust valve

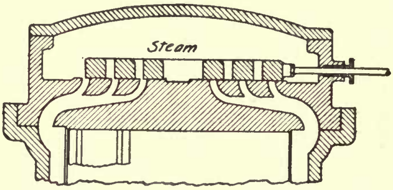


FIG. 83

for each end, it is called a ***three-valve engine.*** When gridiron valves are used, it is more common to have a steam valve and an exhaust valve for each end of the cylinder. The engine then has four valves.

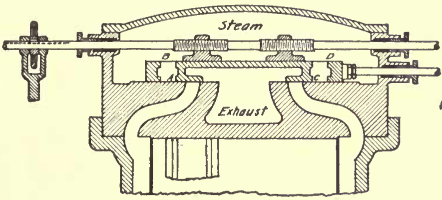
If a governor is so constructed that it regulates the amount of steam admitted to the cylinder by changing cut-off, it is evident from the valve diagrams that the events of release, compression, and admission are changed when cut-off is changed if a single valve is used. Under some conditions this is a serious defect, and it makes the use of separate steam and exhaust valves desirable.

121. The Riding Cut-off Valve. — To utilize the expansive force of the steam in an engine, it is necessary to have an early cut-off. With a single valve, early cut-off will necessitate either early release or early admission. With an early cut-off and with release near the end of the stroke, compression is bound to occur too soon for satisfactory operation under non-condensing conditions. The student only needs to draw a valve diagram to convince himself of this fact. To use an early cut-off, and to have at the same time reasonable percents of release and compression,

a ***riding cut-off valve*** is often used. There are several forms of riding cut-off valves, but we shall describe only one of them.

Figure 84 shows the Myers riding cut-off valve. A main valve slides on the seat in the same manner as an ordinary D valve. The steam lap is made small so that the proper relation exists between the events of admission, release, and compression. If the main valve were acting alone, cut-off would occur very late. To give an early cut-off, a rider valve which controls only the event of cut-off is placed on the back of the main valve. The working edge of the rider valve effects cut-off when it matches with the edges of the main valve at B and at D.

In Fig. 84 both valves are shown in their mid-position. This would not occur normally unless one of the valves were discon-



Meyer. Valve

FIG. 84

nected from its eccentric, but the figure is drawn in this manner to give a clearer idea of the laps. Each valve is driven by its own eccentric. The rider valve is made in two parts and the relative position of the two parts may be changed by revolving the valve stem. One part of the rider valve is secured to the valve stem by a right-hand thread and the other part by a left-hand thread. The hand wheel to the left is arranged so that by turning it the valve stem is rotated and the parts of the valve are brought nearer together or moved farther apart, thereby effecting a change in cut-off. With the parts farther apart, cut-off occurs earlier. When the valves are both in mid-position, as shown in Fig. 84, it is easily seen that the rider valve has *negative* steam lap or steam clearance.

To determine the crank position at cut-off from the valve diagrams, it is necessary to consider the relative motion of the two valves. Figure 85 is the Zeuner analysis for the rider valve. The crank circle and the two eccentric circles are shown by the light lines. The point P_1 is the extremity of the diameter of the

right valve-circle for the main valve, and α_1 is the angle of advance for the main eccentric. The point P_2 is the extremity of the diameter of the right valve-circle, and α_2 is the angle of advance for the rider-valve eccentric. If a number of crank positions be chosen and the displacement of the rider valve *relative* to the main valve be laid off on the crank from the center radially outward, a number of points will be established. Connecting

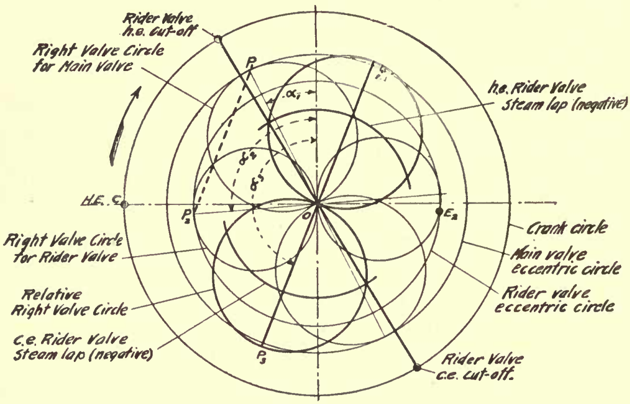


FIG. 85

these points by a smooth curve, we find that it is composed of the two circles shown by heavy lines in Fig. 85. The point P_3 is the extremity of the diameter of the right relative valve-circle and α_3 is the angle which that diameter makes with the vertical.

It is seen that the diameter of the right relative valve-circle, OP_3, is equal in amount and parallel to the dotted line P_1P_2. Knowing this fact, the solution of a rider-valve problem is quite simple, for it is not necessary to plot the points to determine the relative valve-circle.

Since the steam lap for the rider valve is *negative*, the intersection of that lap circle with the *left* valve-circle gives the crank position at head-end cut-off. The cut-off positions of the crank are shown by the heavy lines in Fig. 85. The crank positions for admission, release, and compression are determined from a valve diagram for the main valve in exactly the same manner as previously explained for the D valve.

122. Effect of Rocker Arm on Location of Eccentric. — In the previous discussion of slide-valves, it has been supposed that

the eccentric rod is attached directly to the valve rod, and that the movement of the valve is the same as the horizontal movement of the eccentric. Quite often a ***rocker arm*** is interposed between the eccentric and the valve rods, in which case it may be necessary to modify our previous assumption.

In Fig. 86, three arrangements of rocker arms are shown. In I, the eccentric rod and the valve rod are both connected to the same pin at B, and our assumption is not changed.

In II, the arm BAC reverses the motion of the eccentric. In this case, if a direct valve is used, the eccentric must be placed on the shaft at 180° from the position it would have had without the rocker arm, *i.e.* if a direct valve is used, the eccentric must *follow* the crank by an angle of $90° - \alpha$.

If an indirect valve is used with a reversing rocker, the eccen-

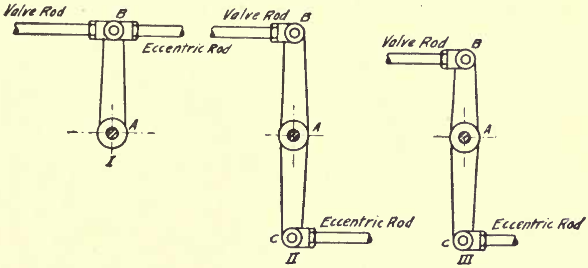


FIG. 86

tric is placed $90° + \alpha$ *ahead* of the crank. This modification does not effect the work in making the valve analysis.

In case III, the two arms of the rocker AB and AC are not of the same length. The travel of the valve is the diameter of the eccentric circle times AB/AC.

123. Oscillating Valves. — One of the most common valves is cylindrical and oscillates or rocks on a cylindrical seat. A spindle fastened to the valve extends out of the steam chest and carries an arm that is moved back and forth by the eccentric. The Corliss valve shown in Fig. 58 and the valves of Figs. 60 and 61 are of this type. It would be possible to make an oscillating valve control both the admission and exhaust events, but this is seldom done. Where the oscillating valve is used, four valves usually are employed. Because of the distortion of motion due to the valve arm, it is not possible to show by the valve diagrams

the exact motion of the valve. However, the relation of horizontal motion of the eccentric still holds, and so the valve diagrams are of value in the analysis of these valves.

124. Poppet Valves. — While they are not common for steam engines in this country, most gasoline engines are equipped with ***poppet valves.*** This is a lifting valve and there is no sliding of the valve on the seat. Figure 87 shows this type of valve. These valves do not have to be lubricated and would seem to be well adapted to conditions where highly superheated steam is used, since one of the troubles met with in the use of superheated steam is the difficulty of proper lubrication of the valve. Where poppet valves are used on steam engines, they usually are operated either by a cam or by an eccentric from a lay shaft that is parallel to the axis of the cylinder. The lay shaft usually is driven by mitre gears from the main shaft.

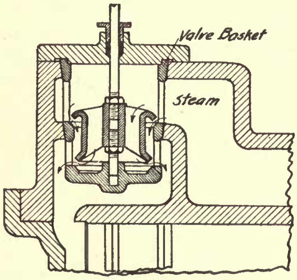


FIG. 87

125. Reversing. — The direction of rotation of an engine may be changed by shifting the eccentric to the proper position. Occasionally an ordinary engine must be reversed. Unless the eccentric is keyed to the shaft, this is not usually a difficult task. With a direct valve, the eccentric leads the crank by an angle of $90°+\alpha$. This is true irrespective of the direction of rotation. To reverse, then, move the eccentric in the direction the engine has been running through an angle of $180°-2\alpha$. The same rule applies with an indirect or inside-admission valve.

With certain classes of engines, such as are used for locomotives, in marine work, etc., reversing is a common occurrence. Some handier and quicker means must then be provided than that mentioned above. The devices used for this purpose are called reversing gears. There are a very great many types of reversing gears in use, but space will not permit the discussion of more than the most common types.

126. The Stephenson Link. — One of the most widely used reversing gears is the ***Stephenson link gear,*** which is much used for small locomotives. In this arrangement, there are two ec-

centrics placed on the shaft at an angle of $180° - 2\alpha$ apart. The forward eccentric controls the forward motion, and the backward eccentric controls the reverse motion. In the diagram of Fig. 88, the crank is shown on head-end dead center, and the valve is indirect, so that the eccentrics will be at an angle of $90° - \alpha$

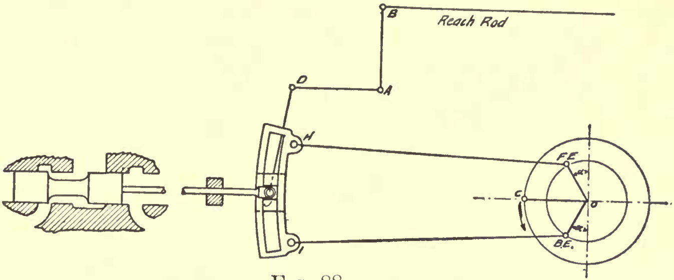


FIG. 88

from the crank. In Fig. 88, *FE* is the forward eccentric, and *BE* the backward eccentric. The two eccentric rods connect to the eyes of a link at *H* and *I*. This link may be raised or lowered by the bell-crank *BAD* and the link *DJ*. When the link is down, as shown in Fig. 89, the forward eccentric entirely controls the motion of the valve. With the link all the way up, the backward eccentric controls the valve. In Fig. 88, the link is

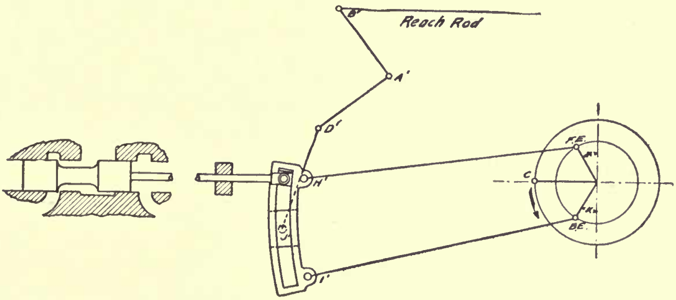


FIG. 89

shown in mid-position with both eccentrics controlling an equal amount. With the crank on head-end dead center as in Fig. 88, the valve is as far to the left as it can get, the port is open a distance equal to the lead, and the valve is at a distance equal to the steam lap plus the lead from its mid-position. With the

valve opening only lead distance, the engine will not get enough steam to run. At mid-gear, half the travel of the valve is equal to the steam lap plus the lead.

With the link in some position between those shown in Figs. 88 and 89, both eccentrics will control the motion of the valve, but the forward one will predominate, and the engine will run forward. The cut-off will now be earlier than it would be if the link was all the way down in full gear. It is evident that the Stephenson gear may be used to change the cut-off as well as to reverse.

It is possible to give the same motion to the valve with the link in intermediate position, by a simple equivalent eccentric. The method of determining this equivalent eccentric is not difficult but it will not be discussed here.

127. The Walschaert Valve Gear. — Most large locomotives in this country are equipped with the ***Walschaert gear,*** or some

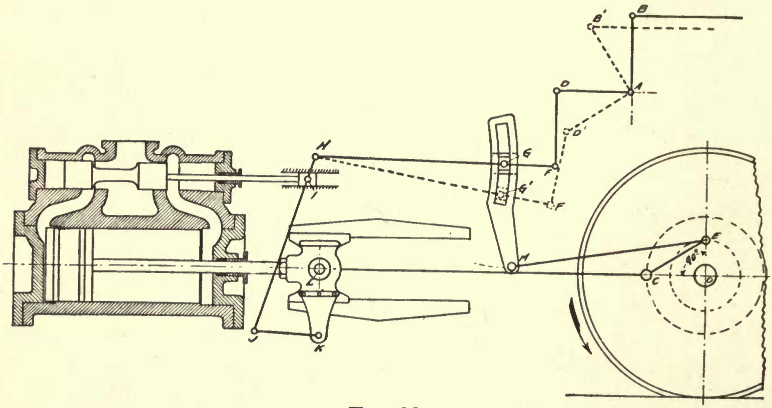


Fig. 90

similar reversing gear. In this type of gear, shown in Fig. 90, the motion of the valve is derived partly from the cross-head, and partly from a return crank or eccentric. That part of the motion coming from the cross-head is constant, while that derived from the eccentric is varied for different conditions of load and direction of rotation. Since the eccentric is placed outside the driver, it is commonly called a return crank. The bar *CE* is fastened to the outer end of the crank pin *C*, and the eccentric pin *E* moves in the dotted circle (Fig. 90), about *O* as a center.

The angle between the crank and the eccentric is 90°. The horizontal motion of the eccentric is transmitted through the eccentric rod EM to the lower point of a link. The link is pivoted at its center G to the frame of the engine, so that the point M oscillates about the point G. In this link is fitted a block which can be raised or lowered in the link by the bell crank DAB. As shown in the dotted position, the block is at its lowest position in the link, and the engine is running forward, taking steam during the largest possible part of stroke. In the full line position, the block is at the center of the link, and the engine will not get enough steam to drive it. With the block in mid-position in the link, the eccentric will give no motion whatever to the valve. Under this condition the motion of the valve comes entirely from the cross-head.

The lever HIJ is called the *combination lever*, because it combines the motion from the eccentric with the motion from the cross-head. The ratio IH/JH is fixed by the condition that

$$\frac{IH}{JH} = \frac{2(\text{steam lap plus lead})}{\text{length of stroke}}.$$

With the block in the center of the link at G, H has no horizontal motion, but the horizontal motion of I is equal to

$$\frac{\text{length of stroke} \times IH}{JH} = 2\ (\text{steam lap plus lead}).$$

As the motion of the valve at mid-gear is 2(steam lap plus lead), it is seen that the valve will open only a distance equal to the lead on each end, and the engine will not get enough steam to run. When the block is dropped to the dotted position, the point H does have a horizontal motion which comes from the eccentric, and with the engine running in the direction shown, the horizontal motion of H will add to the port opening. To reverse, the block is raised above the center of the link.

By changing the position of the block in the link we may get not only a reversal of direction of rotation but also a change in cut-off. As in the Stephenson gear, it is possible to find an equivalent eccentric which would give the motion actually obtained from the mechanism.

128. The Joy Valve Gear. — The *Joy gear* is a so-called *radial gear.* Unlike those previously described, it has no eccentric. Figure 91 shows diagrammatically the principle of its operation. A point such as D on the connecting rod FC will move in the path of an ellipse, which is shown dotted. A bar BED is pinned to the connecting rod at D. The other end of the bar is connected to the frame of the engine by the link AB, A being a point on the engine frame. It is evident that a point E on this bar will have a combination of the elliptic motion of D and the nearly vertical motion of B. The path of E is shown dotted. A bar EGH is connected to BED by the pin E. The point G is in a block which is at liberty to slide along a curved

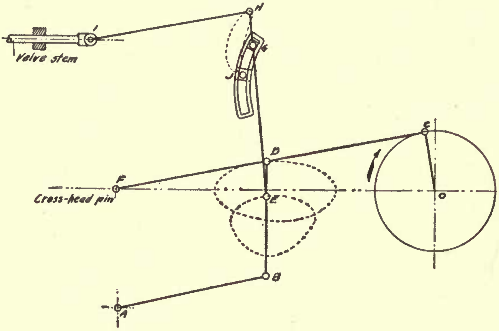


FIG. 91

link. At any particular cut-off this link is held stationary, and the block G slides up and down in it. It is thus seen that the point H gets a combination of the motions of the point E and of the point G. The horizontal component of the motion of H is transmitted through the link IH to the valve. By rotating the link about its center J, a reversal in the direction of rotation of the shaft may be obtained. As in the other gears, cut-off may be varied as well as the engine reversed.

Under conditions of light load an early cut-off may be used and sufficient power obtained if the steam is not throttled between the boiler and the engine. It is the practice on locomotive engines to vary the cut-off to suit the load under normal running conditions. The power generated by the engine might also be regulated by *throttling* the steam, but it has been found that a higher efficiency is obtained at light loads by using an early cut-off

than by using a late cut-off and a low steam pressure, especially where valve gear is used which increases compression when cut-off is shortened, as in the Stephenson gear. In the reversing gears commonly used, an early cut-off is accompanied by an early compression. The increased efficiency at light loads is due, however, more to the early cut-off than to the high compression, although the high compression aids by heating the clearance space, piston, and cylinder head, thereby keeping initial condensation within more reasonable limits.

129. Setting the Slide-valve. — On a small engine that can be turned over easily by hand, the setting of a slide-valve is a simple matter. With proper valve setting an engine should run smoothly, should be easy to start, and each end of the cylinder should furnish about half the power. If an indicator is at hand, it should be employed in the setting. If no indicator is available, the valve may be set by linear measurement. In the setting of a slide-valve there are but two things to do, shift the eccentric on the shaft, and lengthen or shorten the valve stem or rod.

VALVE SETTING BY INDICATOR. In order to get approximately the same amount of work from each end of the cylinder, cut-off for the two ends should be about the same. A rough adjustment may be made easily by placing the eccentric somewhere near its proper position, and adjusting the position of the valve on the rod so that the engine may be started. After the engine is started, take cards and then adjust the length of the valve stem until the cut-off is the same percent on both ends. Next, shift the eccentric until the desired percentage of cut-off is attained. Shifting the eccentric *ahead* makes cut-off come *earlier*. By the use of the indicator it is easy to get the exact setting desired.

Knowing the valve-travel and the dimensions of the valve, we may compute by means of the Zeuner valve diagram the exact amount the valve stem must be lengthened or shortened and the angle the eccentric must be shifted to give a desired setting. In the upper part of Fig. 92 are shown two cards taken with a valve as now set. It is desired to change the setting so that cut-off will be 50 per cent for each end. The cards are shown in length equal to the valve-travel, but they need not have been, as the percentages of stroke could have been scaled and the corresponding crank positions found.

From the cards, locate on the crank circle (assumed for convenience with its diameter the same as that of the valve-travel circle), the crank positions at the different events. Draw lines between the crank positions at admission and cut-off, and from release to compression, for each end. The distance between the admission–cut-off lines should be the sum of the steam laps as measured on the valve itself. A radial line drawn perpendicular to the line joining admission and cut-off establishes the angle of advance α_1. The laps may be measured from the diagram as shown. Now construct a Zeuner diagram (Fig. 93) for the desired

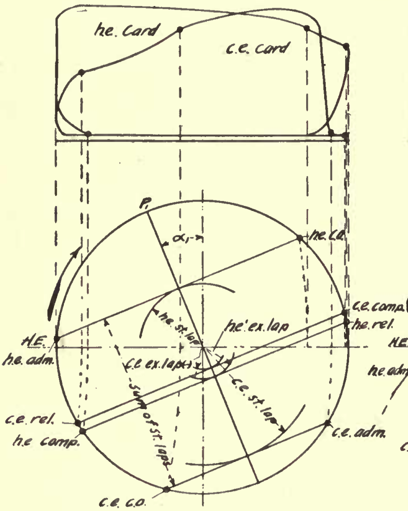


Fig. 92

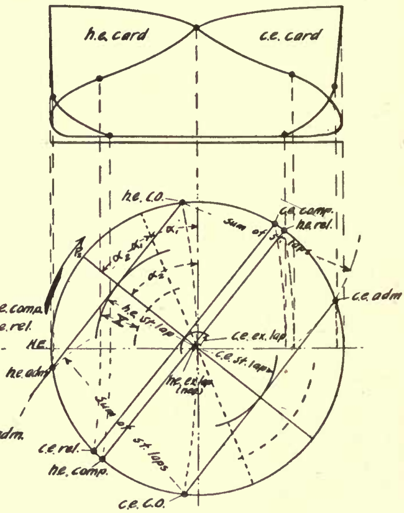


Fig. 93

cut-off (50 per cent in the diagram) keeping the sum of the steam laps the same as before. This amount, and also the sum of the steam lap and the exhaust lap for each end, will be the same no matter what adjustment is made. The angle of advance for the new condition is α_2. Then $\alpha_2 - \alpha_1$ is the angle that the eccentric must be shifted forward. The difference X between the new and the old head-end steam laps is the distance the valve must be moved on the rod. Since the new head-end steam lap is larger than the old, the valve rod must be *lengthened* if the valve is *direct*, and *shortened* if the valve is *indirect*. The upper part of Fig. 93 shows the cards that may be expected after the setting has been changed.

Valve Setting by Measurement. If an indicator is not available, the valve may be set by linear measurement. The steam chest cover must be removed so that the measurements may be taken. It is customary to set either for equal cut-offs or for equal leads. We cannot have equal leads and equal cut-offs at the same setting because of the *angularity of the connecting rod*.* In either case adjust the valve on the stem so that the valve travels about as far beyond the head-end port as beyond the crank-end port.

Setting for Equal Cut-off. By means of marks on the guides and on the cross-head the stroke may be determined, and that proportion from each end at which cut-off is to take place may be laid off on the guides. The procedure is as follows:

(1) Place the cross-head at the position for head-end cut-off.

(2) Loosen the eccentric on the shaft and turn it on the shaft in the direction the engine is to run until the valve is just on the point of cutting off. Fasten the eccentric to the shaft in this position.

(3) Turn the engine to the position for cut-off at the crank-end and measure the distance from the valve to its correct position to give cut-off on this end. Divide the error by two and take up half the error by shifting the valve on the stem and the other half by turning the eccentric on the shaft.

(4) Turn the engine back to the position for head-end cut-off and check the setting. If there is an error left, repeat as explained above.

Be sure the eccentric is fastened firmly to the shaft and the valve to the stem. Replace the cover of the steam chest.

Setting for Equal Lead.

(1) Place the engine accurately on head-end dead center.

(2) Loosen the eccentric on the shaft and turn it in the direction the engine is to run until the port is open a distance equal to the desired lead. Be sure the valve will open the port if the eccentric is turned ahead more. Fasten the eccentric to the shaft.

(3) Turn the engine to crank-end dead center and measure the error. Divide the error by two and take up half by turning the eccentric on the shaft and the other half by adjusting the length of the valve stem.

(4) Turn the engine back to head-end dead center and check. If there is an error, correct it by repeating as previously explained.

* This expression means the deviation from parallelism with the axis of the cylinder, of a connecting rod of finite practical length, except when the crank is at one of the dead centers.

The reason that half the error is taken up by moving the valve on the stem and half by turning the eccentric on the shaft may be explained as follows. Suppose the valve has been set to give the correct cut-off on the crank-end, but that when turned over to head-end position for cut-off there is an error as shown in Fig. 94.

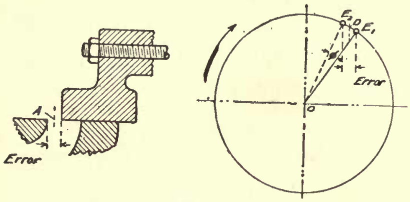


FIG. 94

The eccentric is now at E_1. By turning the eccentric from E_1 back to E_2, this error would be adjusted, but nothing would have been gained as will be seen when the engine is turned back to the position for crank-end cut-off. The same error now exists on crank-end as shown in Fig. 95. That is, the eccentric is at E_4 and should be at E_3. If we try to take up the error by lengthening the valve stem (Fig. 96), nothing is gained because the valve will be moved to the left a distance equal to the error, and when it is turned back to the position for crank-end cut-off, the valve will be open a distance equal to the error. If now we divide the error by two, and move the edge of the valve to A,

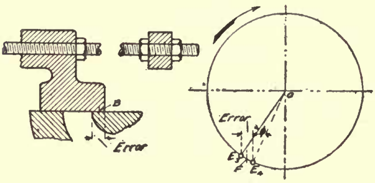


FIG. 95

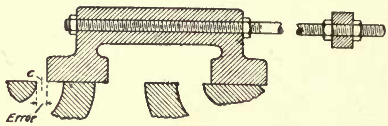


FIG. 96

in Fig. 94, by turning the eccentric back from E_1 to D, we will find that the other edge of the valve will be at B in Fig. 95, when turned over to the crank-end cut-off position. If the other half of the error be taken up in Fig. 94 by lengthening the valve stem *i.e.* by moving the valve to the left on its stem, the valve will be in the correct position for head-end cut-off. Moreover, moving the valve to the left moves the crank-end edge (Fig. 95), from B to the edge of the port, where it should be for crank-end cut-off. Therefore, by taking up half the error in each way at the position of head-end cut-off, we have made no net change in the position of the valve for the crank-end cut-off position.

CHAPTER IX

GOVERNORS

130. General. — The function of the governor is to keep the engine running at nearly the same speed at all loads. It does this by controlling the amount of steam admitted to the cylinder. It is impracticable for the governor to keep the speed exactly constant at all loads. This may be seen when we understand how a governor must work. Under conditions of changing load, the governor must change the amount of steam admitted to do the work. At any instant we have a certain load on the engine, the governor is admitting enough steam to carry that load. If an extra load is thrown on, the engine will momentarily slow down. This slower speed affects the position of the parts of the governor, and this in turn allows more steam to enter to do the extra work required. This change cannot be affected instantaneously, although some governors respond in a very short time. Governors that are quick in making the change are said to be ***sensitive,*** and those that are slow, ***sluggish.***

Under full-load conditions, the engine usually runs slower than that at no load, by an amount that depends upon the construction of the governor. Governors are made that give an engine speed which is nearly as great at full load as at no load. These are said to give ***close regulation.*** It is possible to design and construct a governor that gives the same average engine speed at all loads, in which case the governor is said to be ***isochronous.*** Such a governor would tend to ***hunt,*** *i.e.* there would be a constant fluctuation in speed as the governor attempted to regulate the steam supply to balance the load conditions.

Practical considerations limit the nearness to which isochronism may be approached. If the no-load speed is greater than the full-load speed the governor is said to be ***stable.*** If the governor is isochronous or gives a full-load speed greater than no-load speed, it is ***unstable.*** An unstable governor is clearly undesirable. Various schemes are used to express the relation of speeds at various loads. A common way is to express the variation of speed from no load to full load and from no load or full load to normal load in percent of the speed at normal load. We may

then say, the percent of variation in speed from no load to full load is equal to $100(n_1-n_2)/n$, where n_1 and n_2 are the speed at no load and the speed at full load, respectively, and n is the speed at normal load.

131. Classification of Governors. — Governors may be classified according to the following characteristics.

(*a*) As to the manner of regulating the steam supply: Under this head we have (1) ***throttling governors,*** which regulate the amount of steam admitted to the cylinder by controlling a throttle valve, and (2) ***cut-off governors,*** which control the steam supplied by changing the point of cut-off.

(*b*) As to the predominant controlling force in the mechanism: We speak of ***centrifugal governors, inertia governors*** (although inertia is not a force), and ***resistance governors.*** All mass has inertia. If the mass of the moving parts is small and the inertia effect is not used in governing, we call the governor a *centrifugal governor.* If the inertia is large and its effect is used to aid in governing, we have what we call *inertia governors.* Even in inertia governors, the centrifugal force is a very important factor.

(*c*) As to the force used to balance the centrifugal force of the rotating parts: We have ***gravity-balanced governors*** and ***spring-balanced governors.***

(*d*) As to the arrangement of the mechanism: There are ***spindle governors*** and ***shaft governors.***

132. The Gravity-balanced Spindle Governor. — The diagram of Fig. 97 represents a simple gravity-balanced spindle governor. This is sometimes called a ***conical pendulum,*** and is also often called the ***Watt governor,*** because James Watt first used it on his engines. Two flyballs, at the ends of arms, rotate about a vertical spindle. The arms are pivoted to the spindle at O. The height of the balls is caused to control the steam supply. In Fig. 97 this is done by raising or lowering the point A with the balls, which, by a suitable mechanism, causes either the movement of a throttle valve or a change in the point of cut-off.

A definite relation exists between the height h_1 of the cone of revolution, and the speed of the spindle. To determine this relation consider one of the balls as a free body. At any certain speed it may be considered to be in equilibrium under the action of the following forces: the tension in the arm T, the weight W,

and the centrifugal force acting radially outward $(W/g)\times(v^2/R)$. Taking moments of these forces about the point O, we have

$$\frac{W}{g}\times\frac{v^2}{R}\times h_1 - W\times R = 0.$$

But $v=2\pi Rn$, where n is the number of revolutions per unit time. Therefore we may write

$$\frac{4\pi^2 WR^2n^2h_1}{gR} = WR$$

or

$$h_1 = \frac{g}{4\pi^2n^2}.$$

If we wish to express h_1 in inches and the speed of the governor in r. p. m., our equation becomes

$$h_1 = \frac{32.2\times 12\times 3600}{4\pi^2\,n^2} = \frac{35200}{n^2},\ \textit{approximately.}$$

From this equation it is seen that the height h_1 of the cone of revolution does not depend upon the length of the arm. Figure 97

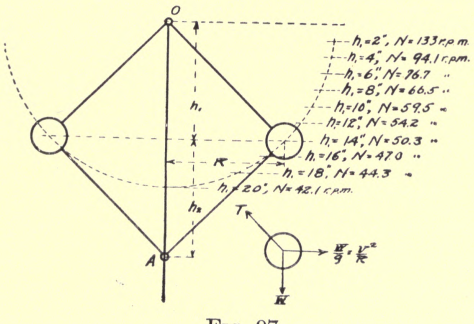


FIG. 97

shows the r. p. m. corresponding to the height h_1 for two-inch increments of h_1 up to 20 inches. From these values it will be noticed that for a certain vertical movement of the ball there will be less speed variation with the ball in the lower positions. In other words, to get a reasonable speed variation it will be necessary to run the governor very slowly. At low speeds the governor will not have much power unless the balls are made excessively heavy. This practical limitation precludes the use of this governor on modern engines.

If the governor arms are crossed, as shown in Fig. 98, it will be noticed that much less variation in speed exists for the same vertical movement of the balls than for the form shown in Fig. 97. Moreover, this governor is nearly isochronous at 50 r. p. m. By the proper selection of the pivots *B* and *C*, we may get quite satisfactory speed regulation for a certain limited range of vertical movement at any desired speed.

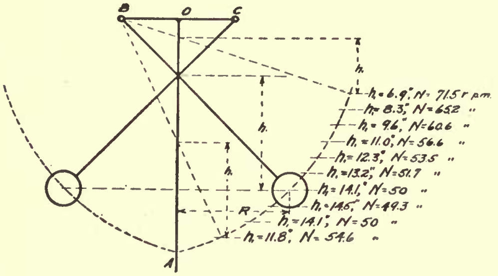


FIG. 98

The pendulum governor may be made exactly isochronous by making the balls swing up in the arc of a parabola, as in Fig. 99. The subnormal of a parabola is constant, and it is seen in Fig. 99 that h_1 is the sub-normal of the parabola which is the path of the balls as they swing upward. The balls may be made to take the parabolic path by having the arms made flexible and to unwind from the evolute of the parabola or by having them guided by an arrangement of cams. Of course it is understood that, in practice, the governor never would be made exactly isochronous, but it is seen that isochronism may be approached as nearly as practical conditions will permit.

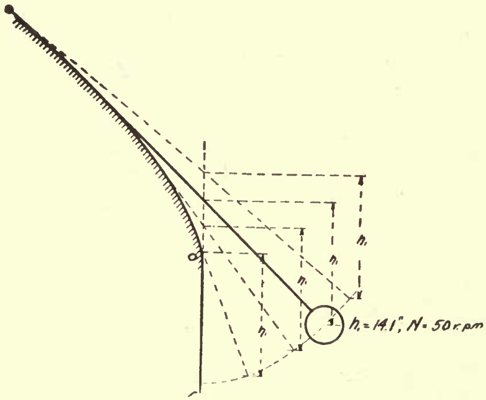


FIG. 99

In order to run a spindle governor at fairly high speed, and still have a reasonably small speed variation in the engine, it is customary to load it as shown in Fig. 100. The load *L* tends to pull the balls down; hence they must rotate faster, to get to the same height as before. To find the relation between the height of the cone of

revolution and the speed, consider the load and the ball each as a free body. With the forces acting on them as shown, we can express the conditions of equilibrium as follows. Expressing the fact that the sum of the vertical forces is zero for the load L,

(1) $$2T_2 \sin \beta = L,$$

or

$$T_2 = \frac{L}{2 \sin \beta}.$$

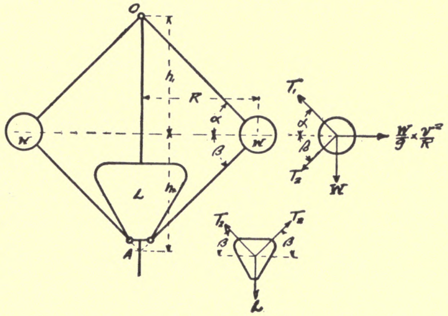


FIG. 100

Considering the ball as a free body, and taking moments about O as a center, we have, since the sum of these moments must be zero,

(2) $$\frac{W}{g}\frac{v^2}{R}\times h_1 - W\times R - T_2 \sin \beta\times R - T_2 \cos \beta\times h_1 = 0.$$

By (1) this may be written in the form

(3) $$\frac{W}{g}\frac{v^2}{R}\times h_1 - WR - \frac{L}{2}\times R - \frac{L}{2} \operatorname{ctn} \beta\times h_1 = 0,$$

or, since $v = 2\pi nR$,

(4) $$\frac{W}{g} 4\pi^2 Rn^2 h_1 - WR - \frac{L}{2} R - \frac{L}{2} \operatorname{ctn} \beta\times h_1 = 0,$$

whence, since $\operatorname{ctn} \beta = R/h_2$,

(5) $$\frac{W}{g} 4\pi^2 Rn^2 h_1 - WR - \frac{RL}{2}\left(\frac{h_2+h_1}{h_2}\right) = 0,$$

and finally, solving for n^2,

(6) $$n^2 = \frac{\left[W + \frac{L}{2}\left(\frac{h_2+h_1}{h_2}\right)\right]}{\left[\frac{W}{g} 4\pi^2 h_1\right]} = \frac{\left[1 + \frac{L}{2W}\left(\frac{h_2+h_1}{h_2}\right)\right]}{4\pi^2 h_1} g.$$

If $h_1 = h_2$

(7) $$n^2 = \frac{\left(\frac{W+L}{W}\right) g}{4\pi^2 h_1}, \quad \text{or} \quad h_1 = \frac{\left(\frac{W+L}{W}\right) g}{4\pi^2 n^2}.$$

If h_1 be expressed in inches and n in r. p. m.,

(8) $$h_1 = \left(\frac{W+L}{W}\right)\frac{35200}{n^2}.$$

The usual form of loaded governor is shown in Fig. 101. This is seen to vary slightly from Fig. 100. Taking the load as a free body, $T_2 = L/(2 \sin \beta)$, as in (1) above.

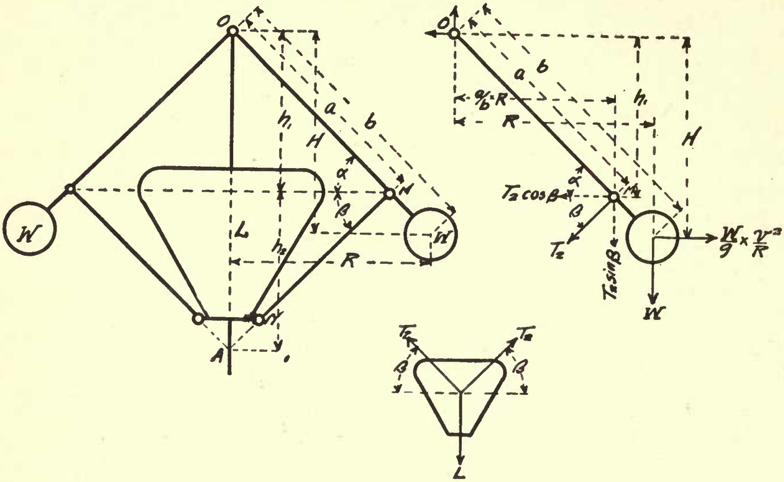


FIG. 101

Taking the upper arm as a free body, and expressing the fact that the sum of the moments about O equal to 0,

$$(9) \quad \frac{W}{g} \times \frac{v^2}{R} \times H - W \times R - T_2 \sin \beta \times \frac{a}{b} R - T_2 \cos \beta \times h_1 = 0.$$

Substituting the value of T_2 from (1), we have

$$(10) \quad \frac{W}{g} \times \frac{v^2}{R} \times H - W \times R - \frac{L}{2} \times \frac{a}{b} \times R - \frac{L}{2} \operatorname{ctn} \beta \times h_1 = 0.$$

But we have

$$(11) \quad h_1 = H \frac{a}{b} \quad \text{and} \quad \operatorname{ctn} \beta = \frac{a}{b} \times \frac{R}{h_2};$$

hence

$$(12) \quad \frac{W}{g} \times \frac{v^2}{R} H - W \times R - \frac{L}{2} \times \frac{a}{b} \times R - \frac{L}{2} \times \frac{a}{b} \times \frac{R}{h_2} \times H \frac{a}{b} = 0.$$

Since we have also $v = 2\pi R n$,

$$(13) \quad \frac{W}{g} 4\pi^2 n^2 H - W - \frac{L}{2} \frac{a}{b} - \frac{LH}{2h_2} \left(\frac{a}{b}\right)^2 = 0,$$

whence

$$(14) \quad n^2 = \frac{\left[W + \frac{L}{2} \frac{a}{b} \left(1 + \frac{H}{h_2} \times \frac{a}{b}\right)\right] g}{W 4\pi^2 H}.$$

Taking as our units inches and r. p. m., this becomes

$$(15)\qquad n^2 = \frac{\left[1+\frac{L}{2W}\times\frac{a}{b}\left(1+\frac{H}{h_2}\times\frac{a}{b}\right)\right]\times 35200}{H}.$$

In the solution of a problem for this type of governor it is best to make a drawing and to scale from it the values of H, h_1, and h_2 at the different positions of the weights. With these values substituted in the formula (15), the r. p. m. of the governor is readily determined. The governor of Fig. 101 is the one commonly used on Corliss engines.

133. The Spring-balanced Governor. — In most high-speed engines, the centrifugal force of the revolving weight is balanced

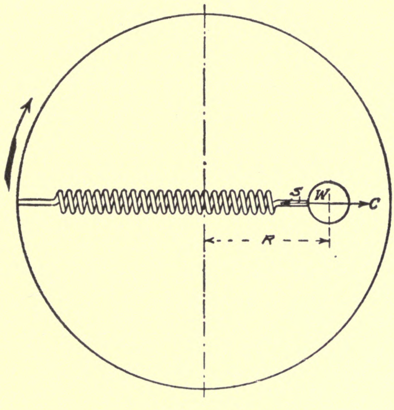


FIG. 102

by the force of a spring. Figure 102 shows a weight W, revolving about the center of a spindle or shaft. The centrifugal force C of the weight is balanced by the spring tension S. The weight is at a distance R from the center of rotation. Hence we have

$$C = \frac{W}{g}\times\frac{v^2}{R} = \frac{W}{g}\,4\pi^2 Rn.$$

For a certain value of n, C varies directly as R. In Fig. 103, this variation is shown graphically. At speed n and with a radius R, the centrifugal force is C. If R is doubled, C is doubled. For other values of n, as n_1 and n_2, C will have different values with the same radius R.

In any kind of uniform spring the elongation, shortening, or deflection is proportional to the force producing the deformation. Figure 104 shows graphically the relation between the pull of the spring and the elongation. If Figs. 103 and 104 be superimposed, as in Fig. 105, we easily can see the relations that must exist if

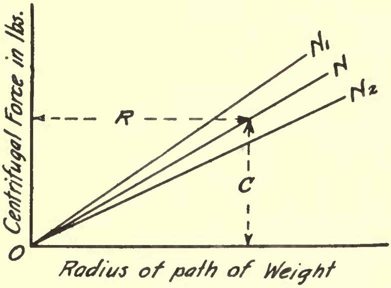


FIG. 103

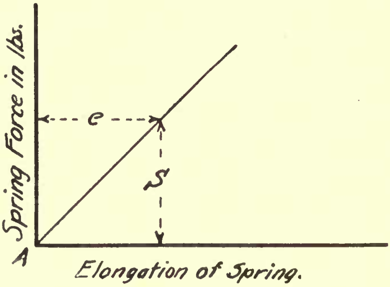


FIG. 104

the spring pull equals the centrifugal force. For the three speeds n_1, n and n_2, the spring pull equals the centrifugal force, as seen at the points b, d, and f. That is, the spring pull will balance the centrifugal force at the speed n_1 when the elongation of the spring is e_1. The weight will then be revolving at a radius $R_1 = a + e$, where a is the distance the weight would be from the center of rotation if there were zero tension in the spring. At a speed n, the forces will balance when the elongation is e and the radius is $R = a + e$. In like manner, the forces balance at the speed n_2 with an elongation e_2 and radius R_2. If the origin A should be moved over to the origin O, *i.e.* the distance a be made zero, it is seen that the spring pull could balance the centrifugal force at only one speed. This means that the governor would then be isochronous. If the origin A were moved to the left of the origin O, the governor would be unstable, because the speed n_1 at no load would be less than the speed n_2 at full load. As stated before, an unstable or isochronous governor could not be used in practice.

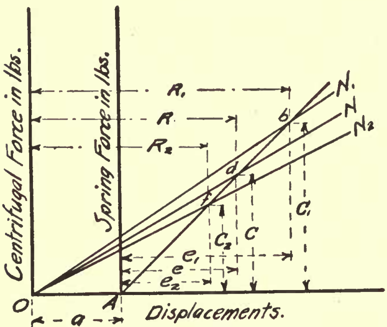


FIG. 105

What is known as ***scale of spring*** is the force necessary to produce an elongation of one inch in the spring. If the spring pull is S_1 at elongation e_1, and S_2 at elongation e_2, the scale of the spring is equal to

$$\frac{S_1-S_2}{e_1-e_2}.$$

Suppose that we desire to find the scale of spring necessary for a governor such as that shown in Fig. 102. Let us suppose that the no-load speed n_1 and the full-load speed n_2, and the corresponding radii R_1 and R_2, are known. First compute C_1 and C_2 for the two speeds. Then, since the spring pull must equal the centrifugal force at all loads, the scale of spring is seen to be

$$\frac{C_1-C_2}{R_1-R_2}$$

because $$R_1-R_2=e_1-e_2.$$

In actual governors it is very seldom that the spring pull acts in the same line as the centrifugal force. Figure 106 represents

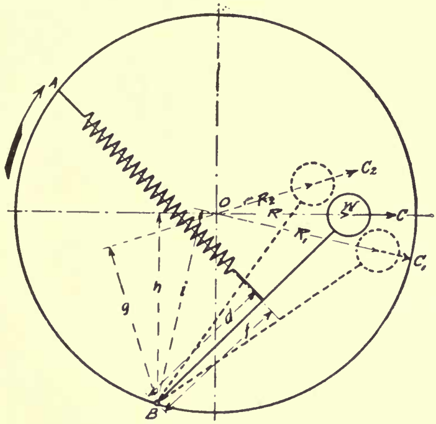


FIG. 106

a more usual case. In the solution of this problem, moments will be taken about the pivot point of the governor arm B. In the full-line position, the moment of the centrifugal force equals the moment of the spring pull about the point B. That is, $C\times h=S\times d$. In like manner, $C_1\times i=S_1\times f$. The scale of spring equals $(S-S_1)$ divided by the elongation of the spring as the weight goes from R to R_1.

134. Governing by Changing Position of Eccentric. — Most shaft governors regulate the steam supply by changing the percent of cut-off. This is accomplished by changing the position of the eccentric relative to the crank. In Fig. 107, a Zeuner valve diagram is shown on which appear two positions of the crank at cut-off. In the full-line construction, cut-off comes late and the angle of advance is α_1, *i.e.* when the crank is on head-end dead center, the eccentric will be at E_1 (direct valve). By shifting the eccentric forward to E_2, the angle of advance is

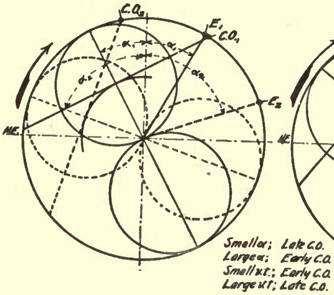


FIG. 107

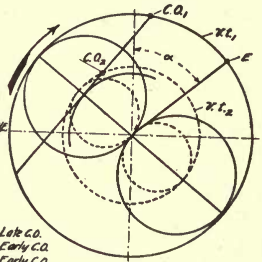


FIG. 108

changed to α_2. The dotted construction gives the position of the crank with the new angle of advance. It is seen that *the cut-off comes earlier with the larger angle of advance.* By turning the eccentric on the shaft, the time of cut-off can be changed but the value of the steam lap will be the same for all values of α, because the only way to change the laps is to move the valve on the stem.

Figure 108 shows the effect on cut-off of changing the valve travel while keeping the angle of advance constant. The Zeuner construction for a large valve-travel is shown in full line. Cut-off is seen to come fairly late. With a smaller valve-travel, as shown by the dotted construction, cut-off is seen to come earlier. Hence it appears that late cut-off is obtained with large valve-travel and earlier cut-off with small valve-travel.

A governor can regulate cut-off by changing either α or the valve-travel. It may be so constructed that it will control by making only the one change or it may change the valve-travel and the angle of advance at the same time.

Changing the angle of advance affects the other events as well as cut-off. When α is increased all events occur sooner. Thus on one-valve engines that control the steam supply by shifting the eccentric, it is found that a high compression accompanies early cut-off, such as is characteristic of the Stephenson valve-gear.

135. Governing by Changing α. — In Fig. 109, a governor is shown that controls cut-off by turning the eccentric around the shaft. The two weight arms are pivoted at the points G and F.

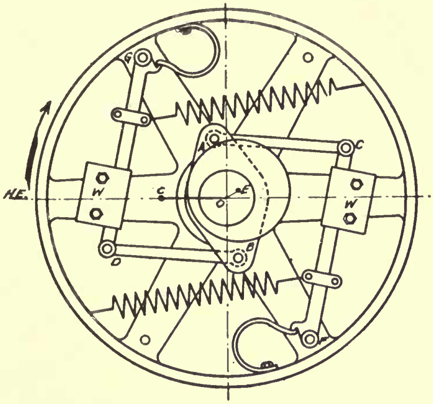


Fig. 109

Any rotation of these arms about their pivots causes the eccentric to turn on the shaft. The weight arms are connected by the links AC and BD to AB, which carries the eccentric. At light loads the speed of the engine is greater than at heavy loads, and the weights will be farther from the center of rotation. This movement of the weights away from the center of rotation causes the eccentric to turn in a clockwise direction relative to the crank. It is evident that this increases α and therefore makes cut-off come earlier, as it should at light loads.

136. Governing by Changing both α and the Valve-travel. — In Fig. 110 a governor is shown that changes both α and the valve-travel at the same time. The pivot point on the flywheel carries the governor arm. The eccentric is shown as a pin. When the governor arm moves about the pivot point in a clockwise direction, it carries the center of the eccentric with it and

makes α smaller and the valve-travel larger. It is known that small α and large valve-travel both give late cut-off. Conversely, a counter-clockwise movement of the arm about the pivot point gives early cut-off because it makes α larger and the valve-travel smaller. The centrifugal force acts through the center of rotation and the center of gravity. Hence this force tends to give the arm counter-clockwise movement about the pivot. At light loads, and therefore higher speeds, the centrifugal force will be greater than for heavy loads. This tends to give the arm counter-clockwise rotation about the pivot and this makes cut-off come

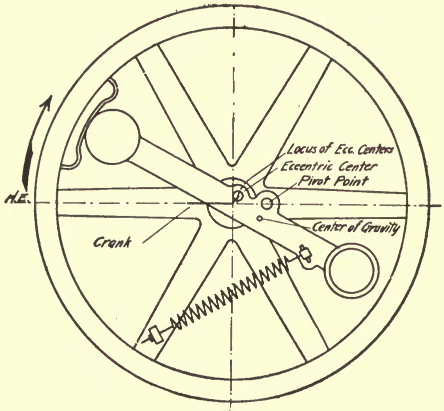


Fig. 110

earlier for light loads, as it should. A great many other governors besides the one shown in Fig. 110, change α and the valve-travel in the same way.

137. Centrifugal and Inertia Governors. — As has been previously stated, all governor weights have inertia. If the tendency of the weight to keep moving at the same speed helps to effect the change of position that causes the governing, the governor will respond more quickly than it would if the inertia opposed the change. In the gravity-balanced spindle governors that were considered in §132, inertia acts against the rapid change of position of the balls and so tends to make the governor sluggish.

In the governor of Fig. 110, the inertia of the arm assists in the governing. If the load is suddenly thrown off the engine,

it will momentarily speed up. This means that the flywheel will go ahead of the governor arm or rotate in a clockwise direction relative to the arm. This swings the eccentric nearer the center, which makes α large and the valve-travel small. Hence cut-off occurs sooner, which tends to re-establish the conditions of equilibrium. In Fig. 110 the arm is made very heavy, so that it will have considerable inertia.

It must not be assumed that the inertia of the arm is the only factor in the governing. The centrifugal force also plays its part, as has been explained previously. The governor of Fig. 110, while it is very simple in construction, is at the same time sensitive and quick in its action. It is widely used and is known as the ***Rites inertia governor.***

CHAPTER X

STEAM TURBINES

138. Introduction.—The steam turbine of to-day is of as much importance in the world of engineering as is the reciprocating steam engine. Practically all large steam power plants which produce electric current employ the turbine engine. The development of the turbine has been remarkably rapid. However, it would not be true to say that the turbine has crowded the reciprocating engine from the power-plant field. The fact is rather that it has developed along new and different lines of use, and now occupies a field that was never held by the reciprocating engine, *i.e.* as the direct connected prime mover for high-speed electric generating units.

The turbine and the alternating-current generator have developed together. Both the turbine and the alternating-current generator are well adapted to high-speed rotation. The cost of a slow-speed electric generator is much higher than that of a high-speed generator of the same capacity. Before the days of the turbine, generators were nearly all of the direct-current type, which could not run at very high speeds.

The electric system of power transmission is more economical than the old belt system. Hence the turbine has replaced the reciprocating engine in some manufacturing plants on account of the development of systems of electric power transmission. On land, large turbines are seldom used to drive anything but electric generators.

139. History. — The steam turbine is not a modern invention. Hundreds of years ago people knew, as every child knows today, that a pin-wheel would rotate when blown upon. There are records of turbines built in the quite distant past. They were little but toys, however, like pin-wheels, and of no practical importance. The modern turbine dates from the years between 1880 and 1890. During this period two types of turbines that have become of great practical importance were developed.

De Laval, the inventor of the cream separator, sought to drive his separator by means of a turbine. After several experiments, he perfected a type that was satisfactory for that purpose. The

same turbine, with improvements, has been used in large numbers for driving centrifugal pumps, fans, and even small generators.

During practically the same time, C. A. PARSONS developed the type of turbine that now bears his name. These two pioneers were soon followed by other experimenters. Various forms of turbines were developed. Some of these are still used. Others are obsolete and are of interest only from an historical standpoint.

In the development of the turbine, there were two obstacles that had to be overcome. The first of these was the lack of knowledge of principles. The second was the need of better mechanical means of manufacture. As will be shown later, the velocity of the rotor in a turbine must be very high. This causes large stresses, and makes necessary a very perfect balance. The clearances between the rotors and the stationary parts must be small to prevent undue leakage. This calls for an excellence of design and construction that did not commonly exist for heavy machines in the past. As in the development of any new machine, satisfactory solutions of the problems grew out of the necessities, so that the modern turbine is as reliable and dependable as any piece of machinery in the power plant.

140. Fundamental Principles. — Before making a study of the common types of turbines now in use, we shall discuss the fundamental principles of the steam turbine. It is not our purpose to give an exhaustive discussion, but only to present the principles in their simplest form. The sketches of blades and nozzles are not exactly correct in shape for the conditions assumed. They are to be considered as only diagrammatic.

Steam under pressure contains a certain amount of usable heat. The available amount depends upon the initial pressure, the degree of superheat, and the pressure to which the steam may be dropped. There is the same amount of available heat if the initial and final conditions of the steam are the same, no matter whether we are considering the reciprocating steam engine or the steam turbine. The turbine, or the reciprocating engine, is efficient, or is not, according as it uses a large or a small amount of this available energy.

Since the turbine and the reciprocating engine both use the same medium, it is not to be expected that one will be much more efficient than the other. Both reciprocating engines and

turbines may be made of about the same thermal efficiency. The choice of type of engine depends upon other considerations than efficiency.

The greatest loss in the reciprocating engine is due to the initial condensation in the cylinder. Since the cylinder walls are made of a heat-conducting material, they will never be as hot as the incoming high-pressure steam, and they will be hotter than the low-pressure steam leaving the cylinder. The relatively hot steam coming to the cylinder strikes the cooler cylinder walls and some condensation takes place, with a consequent shrinkage in the volume. The condensed steam is mostly re-evaporated before the steam leaves the cylinder, owing to an absorption of heat from the then hotter cylinder walls.

The loss in the turbine is due to other causes, such as leakage, friction, etc. The leakage occurs around the ends of the blades or from stage to stage. The friction exists between the steam and the parts of the turbine in the passage of steam through both the stationary and the moving parts. There is also a windage loss between the moving parts and the steam. This friction does not cause a complete loss, because a part of the heat generated may be used in later stages of the turbine.

141. Available Energy in Steam. — In order to make clear the nature of the available energy in steam, a concrete example will be taken.

(1) Let us assume that the steam is dry saturated steam at a pressure of 150 pounds gage (165 pounds absolute), and that it is allowed to expand adiabatically to a pressure of 15 pounds absolute.* The heat contents of a pound of dry saturated steam at 165 pounds absolute is 1194 B.t.u. The heat contents of a pound of steam at 15 pounds absolute, after expanding adiabatically from 165 pounds, is 1019 B.t.u. The difference between these values, which is the amount of heat available for doing work, is $1194-1019=175$ B.t.u. At 165 pounds pressure, dry saturated steam occupies a volume of 2.75 cubic feet per pound. At 15 pounds pressure, after the expansion just mentioned, the quality is 87 per cent, and the volume is about 23 cubic feet.

In the reciprocating steam engine, this change in volume, working by its pressure, does work on the piston in forcing it forward.

* Adiabatic expansion is that in which no heat is added to the steam and none is extracted except by the conversion of heat into work.

The velocity of the piston is immaterial. In the turbine, the same change in volume takes place, but the steam is allowed to acquire velocity in expanding. The energy of the steam due to its velocity is imparted to the rotor of the turbine. The efficiency of a perfect engine working on the Rankine cycle,* between the pressures of 165 and 15 pounds absolute, is $175/(1194-181)=17$ per cent. Since neither the reciprocating engine nor the turbine is perfect, neither would have an efficiency as great as 17 per cent when worked between the pressure limits named.

(2) Assume that the steam is allowed to expand adiabatically from 165 pounds absolute to 1 pound absolute. The heat-drop is $1194-871=323$ B.t.u. and the efficiency on the Rankine cycle is $323/(1194-70) = 28.6$ per cent.

142. Velocity Due to Expansion. — Let us next compute the velocity of the steam if all the heat-drop goes to giving the steam velocity.

(1) Suppose that the drop in pressure is from 165 to 15 pounds absolute. Since one B.t.u. = 778 foot-pounds, the energy is $175 \times 778 = 136{,}100$ foot-pounds per pound of steam. The energy of motion, or kinetic energy, is $mv^2/2 = (1/32) \times v^2/2$. This must be equal to the value 136,100 foot-pounds just calculated. Hence

$$v^2 = 64 \times 136{,}100 = 8{,}710{,}400,$$
$$v = 2950 \text{ ft./sec.}$$

(2) If the drop in pressure is from 165 to 1 pound, we find, in a similar manner,

$$K.E. = 778 \times 323 = (1/32) \times v^2/2,$$

whence

$$v = 4010 \text{ ft./sec.}$$

In the steam turbine, the steam must be expanded, and the velocity due to this expansion must be used by imparting its kinetic energy to the rotor of the turbine. If this is done by allowing the steam to expand in the stationary parts of the turbine and imparting the velocity thus produced to the moving parts, the turbine is said to be of the ***impulse type.*** If it is done

* To compare steam engines, the efficiencies based on the Rankine cycle are often used. The efficiency of a steam engine operating on the Rankine cycle is given by the expression $(Q_1 - Q_2/Q_1$, where Q_1 is the amount of heat required to make dry steam at boiler pressure from water at the temperature of the exhaust, and Q_2 is the amount of heat rejected from the engine minus the heat of the liquid at the temperature of the exhaust.

by allowing the steam to expand in the moving parts, the unbalanced steam pressure reacting on the rotor, the turbine is called a ***reaction turbine.*** In some turbines the steam expands both in the stationary parts and in moving parts, and the turbine is said to be of the ***impure reaction*** type.

143. Impulse and Reaction. — In order to understand the principles involved, consider the simplest cases involving the principles of impulse and reaction in which the velocity is created and used. It may be easier to think of the jet as a jet of water, for in that case the fluid does not expand when the pressure on it is reduced. Otherwise, the steam jet and the water jet follow the same laws of impulse and reaction.

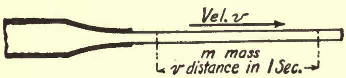


Fig. 111

Suppose that water issues from a nozzle, as in Fig. 111. The nozzle is stationary, and the issuing jet has a velocity of v feet per second. The unit mass m of water that will be considered is that issuing from the nozzle in one second. A particle of water in the jet will move v feet in one second. The kinetic energy of this unit mass will be $mv^2/2$.

(1) Consider the case in which the jet strikes a *stationary flat surface* (Fig. 112). After striking the flat surface, the water flows or splatters out to the sides at right angles to its former direction of motion, that is it loses all its velocity in the direction of the jet. The force exerted by the jet on the flat surface may be measured by the force F necessary to hold the flat surface stationary. Since force = mass$\times$change in velocity per second, and since the time in which mass m emerges from the nozzle and strikes the plate is one second, we have $F = mv$. The force exists in the case of the stationary plate, but no work is done because the plate does not move.

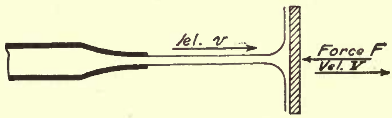


Fig. 112

(2) Suppose that the flat surface moves with a velocity V (Fig. 112). Then the force $F = m\times(v - V)$. The quantity $(v - V)$ is the velocity of the jet *relative* to the flat surface. It is seen that F is less than before, and will be zero if the velocity of the surface is the same as the velocity of the jet. The work done in one

second by the jet on the plate equals F times the distance the plate moves, or

$$W = F \times V = m(v - V)V.$$

The velocity of the plate at which the work is a maximum may be found by equating to zero the first derivative of the work with respect to V. This gives

$$\frac{dW}{dV} = m(v - 2V) = 0.$$

Hence the maximum work occurs when $V = v/2$, that is, when the velocity of the plate is half that of the jet.

(3) If instead of striking a flat surface, the jet strikes a stationary curved surface, such that the jet is turned completely back on itself, or through an angle of 180° (Fig. 113), the force F exerted on the surface is $m \times 2v$, which is twice that exerted on the flat surface. Since the curved surface is stationary, there is no work done.

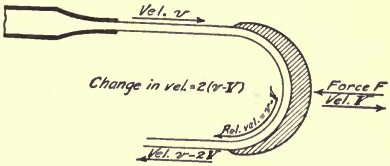


FIG. 113

(4) Suppose the curved surface (Fig. 113) is moving with a velocity V. The velocity of the jet *relative* to the curved surface is $(v - V)$. It follows that the *absolute* velocity of the jet leaving the surface is $(v - V) - V = (v - 2V)$. Consequently the change in the velocity of the jet is $v + (v - 2V) = 2(v - V)$, because the direction of motion is completely reversed. As before, we have

$$F = m \times 2(v - V),$$

and the work done per second is

$$W = FV = 2m(v - V)V = 2m(vV - V^2),$$

whence

$$\frac{dW}{dV} = 2m(v - 2V).$$

For maximum work we must have

$$\frac{dW}{dV} = 0.$$

Hence

$$v = 2V, \text{ or } V = v/2.$$

That is to say, *the curved surface should move at half the velocity of the jet for the production of maximum work.* If the latter condi-

tion exists, the absolute velocity of the jet as it leaves the surface is zero. That is, all of the velocity of the jet has been used.

(5) In Fig. 114 we have a tank that is free to move horizontally upon a track. In one side of this tank is placed an orifice or nozzle. The water issues from this nozzle due to the pressure of the water from above. If the tank is stationary, the water leaves the tank with an absolute velocity v. The force F, due to the unbalanced pressure of the water in the tank, tends to force the tank to the left, but since the tank is held stationary, the force F does no work. If the tank is allowed to move to the left with a velocity V, however, the work done will be FV. The absolute velocity of water leaving the tank is $(v-V)$. The maximum work will be done when $V=v$, that is when the escaping water has no absolute velocity.

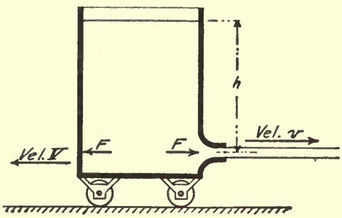


Fig. 114

In the first four cases considered, the jet impinged on a surface. Work was done by the jet striking and moving the surface. A turbine in which the pressure-drop occurs in a stationary nozzle or part is said to be an impulse turbine (§142) because the energy is given to the moving parts by the impulse of the jet. In the fifth case the drop in pressure occurred in the moving nozzle. When this occurs in a turbine, it is said to be a reaction turbine (§142). A comparison of the above simple examples shows that the velocity of the moving parts of a reaction turbine must be nearly twice as great as that of the impulse type, other factors being equal.

144. Bucket Shapes. — In the common types of steam turbines, buckets or blades are mounted on the periphery of a wheel or rotor. The shape of these blades is something like that of the curved surface considered in (3), §143. Of necessity the jet cannot be completely turned through an angle of 180° as in (3), §143, because the steam must have a velocity in the direction of the axis of rotation of the rotor in order to get to the bucket and to leave it.

In Fig. 115, let v_1 denote the velocity of the jet relative to the bucket or blade at the point where the jet first strikes it, and let α

denote the angle it makes with the tangent to the rim of the rotor. Let v_2 and β, respectively, denote the velocity and the angle upon leaving the bucket. We see that the component of the velocity of the jet relative to the bucket in the direction of the tangent is $v_1 \cos \alpha$ and the component in the direction of the axis of the rotor is $v_1 \sin \alpha$. In like manner, the relative velocity v_2 has similar components $v_2 \cos \beta$ and $v_2 \sin \beta$. These components $v_1 \sin \alpha$ and $v_2 \sin \beta$ must be large enough to get the jet through the row of buckets on the rotor, in order that the following buckets shall not interfere with the flow.

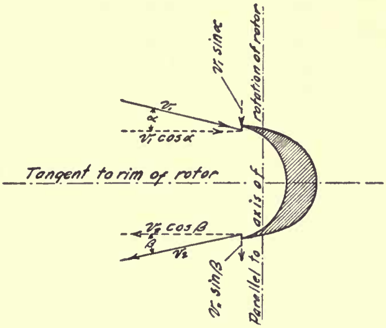


FIG. 115

If the relative velocity of the jet is v_1 and the angle that it makes with the tangent is α, the absolute velocity v of the jet makes a different angle θ with the tangent (Fig. 116). If V denotes the tangential velocity of the bucket, v is the resultant of the two velocities v_1 and V, and θ is the angle that this resultant v makes with the tangent.

In like manner, the resultant of v_2 and V at the exit is the absolute velocity v' at the exit, and it makes an angle ϕ with the tangent.

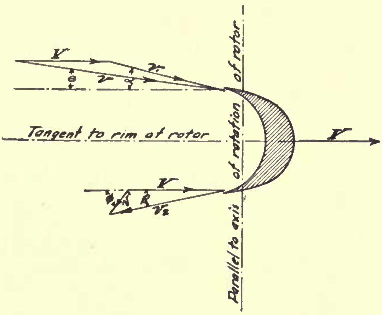


FIG. 116

If we assume that the jet strikes the bucket in the direction of the tangent to the rim of the rotor, the preceding paragraph shows that an error will be introduced. In order to make the calculations as simple as possible, however, we shall assume that the jet does strike tangentially, and we shall bear in mind that some error has been introduced. The results previously derived for steam velocities for certain heat drops will now be applied to the problem of the turbine.

145. The Single-stage Turbine. — In a single-stage impulse turbine, the steam is expanded in a stationary nozzle, and is directed against the moving buckets or blades, which are mounted on the rim of the rotor. In the single-stage reaction turbine, the rotor itself carries the nozzles, and the steam expands in passing through them.

If the expansion of the steam is from 165 to 15 pounds absolute we have seen in (1), § 142, that the steam or jet velocity is 2950 feet per second. For maximum work done, the bucket velocity of the impulse turbine is half that of the jet velocity (§ 143). Hence the peripheral velocity of the rotor should be $2950/2=1475$ feet per second. With a reaction turbine, the peripheral velocity of the rotor should be the same as the jet velocity, or 2950 feet per second.

If the rotor speed is assumed to be 3000 revolutions per minute, or 50 revolutions per second, the diameter of the rotor should be

$$\frac{1475}{50\pi}=9.4 \text{ feet}$$

for an impulse wheel. This is obviously very much too large. For a reaction wheel, the diameter should be 18.7 feet, which is absurd. If a speed of 24,000 r. p. m. is assumed, the diameter of the rotor for an impulse turbine should be

$$\frac{1475}{400\pi}=1.17 \text{ feet,}$$

or 14 inches. These values for the speed and the diameter of the rotor are not far from those which are used in the DeLaval single-stage turbine.

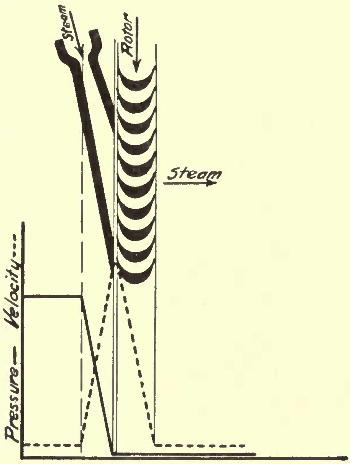
The preceding examples show what a very high peripheral velocity is necessary for a fair efficiency in a single-stage turbine. With a vacuum, a much larger velocity should be used. Since immense stresses are induced in the wheel by these high velocities, it is readily seen that a single-stage reaction turbine is almost out of the question. If such turbines were operated, their efficiency would necessarily be very low. Hence they are not used.

In Fig. 117, a diagram of the single-stage impulse turbine is shown. Steam enters the nozzle from the left, expanding as it passes through. As the pressure drops, a high velocity is imparted to the steam. The steam leaves the nozzle at low pressure and at a high velocity. The steam now impinges upon the

buckets or blades of the rotor, imparting to the rotor its velocity, and therefore its kinetic energy. Upon leaving the rotor, the absolute velocity of the steam is quite low.

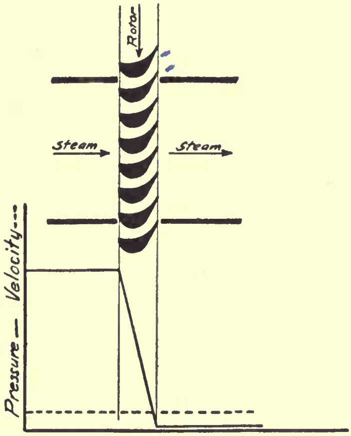
The graphs at the lower part of the diagram show the changes in the pressure and in the velocity. The steam pressure is shown by the full line, and the steam velocity by the dotted line. While single-stage impulse turbines are widely used, they are never made in large sizes.

The diagram of Fig. 118 represents a single-stage reaction turbine. The steam passes from the left directly to the rotor. The rotor carries blades so shaped that the spaces between them act as nozzles. The steam expands in these spaces or nozzles. As it expands, its pressure drops, and it reacts upon the blades. This



Single-Stage Impulse (De Laval)

FIG. 117



Single-Stage Reaction

FIG. 118

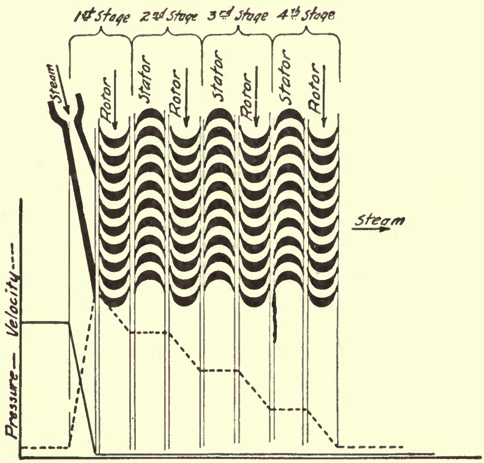
force of the steam on the blades causes the rotor to move and to absorb the energy liberated by the expansion. It will be noticed from the graphs that the velocity does not change much in passing through the rotor, but the pressure drops during the passage.

In order to decrease the peripheral velocity of the rotor, and at the same time to expand and use all the velocity of the steam, more than one set of rotor blades or buckets are employed. This is called ***staging.*** The steam passes successively through the sets of blades in each stage, giving up part of the energy to each set.

146. Staging. — In multi-stage impulse turbines, two methods are in use. The first method is to expand the steam in one set of stationary nozzles, and to take out part of the velocity in each stage. This is known as ***velocity-staging.***

The second method is to expand the steam partially in one set of stationary nozzles, using up the velocity caused by this expansion in one stage, then to expand the steam again in another set of stationary nozzles, using the velocity thus generated in the second stage, and so on. This scheme is called ***pressure-staging.*** A combination of pressure-staging and velocity-staging is also used, in which there are two or more velocity stages in each pressure stage.

147. Multi-stage Impulse Type with Velocity-staging. — The diagram in Fig. 119 shows the velocity-stage impulse turbine.



Multi-Velocity-Stage Impulse.

FIG. 119

The steam enters from the left and passes through the stationary expanding nozzle, where the pressure drops and the velocity is acquired in exactly the same manner as in the single-stage impulse turbine of Fig. 117. The rotor in this case, however, has much less velocity than the rotor shown in Fig. 117. Hence the steam loses only a part of its velocity in passing through the first

set of buckets. Emerging from the first set of buckets, it passes through a set of stationary blades or vanes which change the direction of flow of the steam, but not its velocity. These stationary blades are necessary, because the steam has a large upward component of velocity after leaving the rotating buckets of the first stage; and since the velocity of the rotor is downward, the direction of flow must be reversed so that the steam may impinge on the second set of rotating buckets. In passing through the second set of moving buckets, more of the steam velocity is taken up by the rotor. The direction of flow is again changed by the stator, and so on, till the steam finally emerges from the turbine with its velocity practically all expended.

Suppose, for example, that the downward velocity of the steam as it leaves the nozzle is 4000 feet per second, and that the bucket velocity downward is 500 feet per second. As it leaves the first set of buckets on the rotor, the steam will have an upward velocity of $4000-2\times500=3000$ feet per second, the effect of friction being neglected. In going through the first set of stator blades, the direction of flow is reversed, but is unchanged in magnitude. Upon leaving the second set of rotor buckets its velocity will be upward, and its magnitude will be $3000-2\times500=2000$ feet per second, and so on for the other two stages. Each set of moving buckets takes out 1000 feet per second of its velocity, and it emerges with no vertical component of velocity.

It was shown in §145 that the rotor of a single-stage turbine has to have an absurdly large diameter unless it has a very high speed or the efficiency is very low. The objection to such high speed is that the turbine must have a reducing gear in order that the power may be used. With a multi-stage turbine we can choose the diameter and also the speed, and make the number of stages such that all the velocity can be used.

Let us assume that the speed is 3000 r. p. m., and that the diameter of the rotor is 3 feet. Then the peripheral velocity must be $(3000/60)\times3\pi=471$ feet per second. Neglecting the effect of friction, each stage will absorb a steam velocity of twice the bucket velocity, or 942 feet per second. If the steam expands from 165 to 15 pounds absolute, the steam velocity is 2950 feet per second. Hence the number of stages necessary will be $2950/942=3.1$, and three stages should be used. If the turbine is condensing, and the pressure drops from 165 to 1 pound absolute,

the steam velocity will be 4010 feet per second, the number of stages 4010/942 = 4.2, and four stages should be used.

In a velocity-stage turbine, the efficiency is very low after the first two stages, principally because the jet is broken up by its passage through the blades. As a result, more than two velocity stages are seldom used. It must be remembered also that a very high steam velocity produces a very great friction between the steam and the surfaces of the blades, thereby causing a considerable loss.

148. Multi-stage Impulse Type with Pressure-staging. — If, instead of expanding the steam completely in one nozzle, we expand it only a little in the first nozzle, then use its velocity, expand it some more in a second nozzle, and again use the velocity generated, and so on, the process is called *pressure-staging* (§146). Figure 120 shows the method diagrammatically. In this diagram there are five sets of nozzles and five pressure stages. The steam enters from the left and passes through the stationary nozzle. The pressure and velocity lines below show that the drop in pressure is accompanied by an increase in velocity. The steam with its acquired velocity impinges on the blades of the rotor. The velocity is absorbed in the rotor. As the steam leaves this rotor with low velocity, it is collected and led to a second stationary nozzle in which the pressure is again dropped, and velocity is acquired. The second rotor absorbs this velocity and the steam passes on through the following stages, until its pressure and velocity are practically all used up at its exit.

Making the same assumptions for speed and diameter of the rotor as in the preceding type, let us compute the number of stages necessary with the pressure-stage type. If the diameter of the rotor is 3 feet and the speed is 3000 r. p. m., there is a steam velocity of 942 feet per second to be absorbed per stage (§147). If a pound of steam loses a velocity of 942 feet per second, it gives up

$$\frac{1}{2}\times\frac{1}{32}\times\overline{942}^2 = 13900 \text{ foot-pounds}$$

of kinetic energy. This is equivalent to 13,900/778 = 17.9 B.t.u. For the non-condensing condition assumed in §147, there was a

ouch $A_2 = A_0 + \frac{V_0}{V_2} + \frac{V_2}{V_0} + \frac{X_2}{X_0}$

Velocity

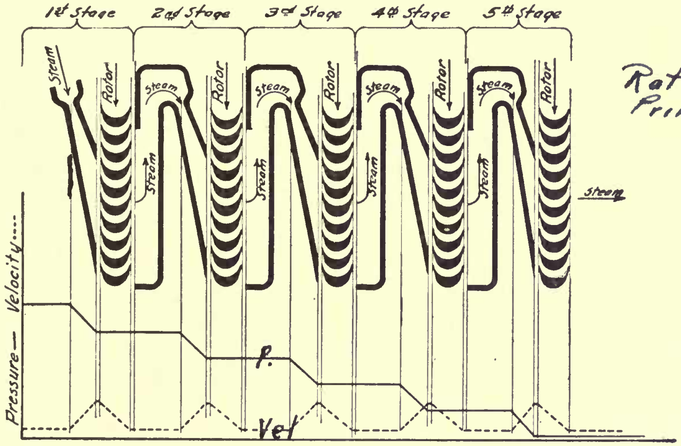
heat-drop of 175 B.t.u. available in the whole turbine. The number of stages will then be

$$\frac{175}{17.9}=10.$$

For the condensing conditions assumed in §147, the number of stages will be

$$\frac{323}{17.9}=18.$$

In making a comparison of this type with that of §147, it might seem at first sight that the velocity-staging were the better, as the number of stages is so much smaller. While this is an advantage, it is overbalanced by the fact that the pressure-stage type is more efficient. This type is used very extensively in



Rateau. Principle

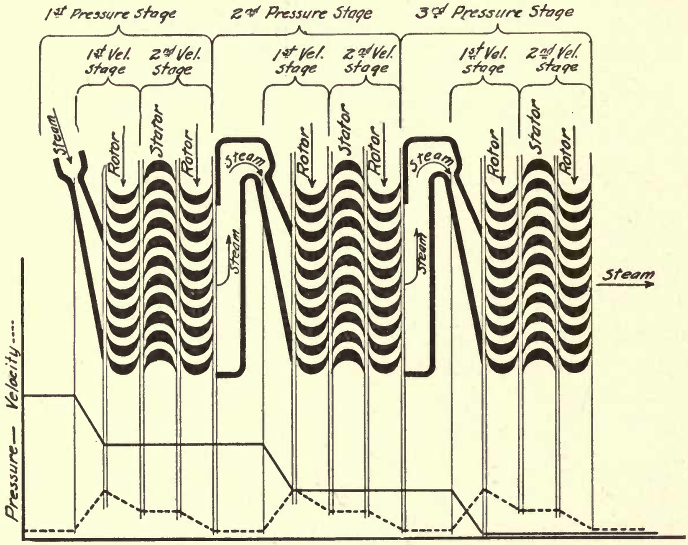
Multi- Pressure-Stage Impulse.

Fig. 120

medium-sized turbines. It is often called the multi-cellular type because each pressure stage is composed of a cell that is steam-tight except for the openings through the inlet and outlet nozzles. It is evident that it is necessary to keep each cell steam-tight in order to prevent a leakage of steam from one stage to the next. Where no difference in pressure exists, there is no tendency to leak. In the velocity-stage type, the pressure was the same throughout the whole turbine beyond the expanding nozzles, and so there could be no leakage.

of rotor to half. of that of steam

149. Multi-stage Impulse Type with Combined Pressure-staging and Velocity-staging. — Very often a combination of pressure-staging and velocity-staging is used, with the result that some of the advantages of both types are utilized. The diagram of Fig. 121 shows this arrangement. In the sketch three pressure stages are shown, and each pressure stage has two velocity stages. The drop in pressure occurs in the three stationary nozzles. After expanding in the nozzle, the steam passes through the buckets



Multi-Pressure, Multi-Velocity Stage (Curtis)

FIG. 121

of the rotor wheel, where a part of the velocity is absorbed. It then emerges from this rotor, and its direction of flow is reversed in the stationary vanes, as in the velocity-stage impulse type. It then passes through a second set of moving buckets, where most of the remaining velocity is absorbed. The steam is now collected and expanded some more in the next nozzle, and the process of the first pressure stage is repeated.

By dropping the pressure in three successive nozzles, the velocity generated is not nearly so great as it is in the velocity-stage type of Fig. 119. This means that there will be less friction loss due to excessively high steam velocities. Ordinarily two velocity

$V_0 = \sqrt{2gn} H \times 778.$

stages are used for each pressure stage, so that the drop in efficiency due to a disturbance of the jet is not so great as in the pure velocity-stage type.

Applying the above problem to this combination type, let us determine the number of pressure stages required. Assuming, as before, that the speed is 3000 r. p. m., and that the diameter is 3 feet, we find that the bucket velocity is 471 feet per second (§147). Since there are two velocity stages for each pressure stage, it is seen that $2 \times 2 \times 471 = 1884$ feet per second of steam velocity per pressure stage can be absorbed. The heat-drop per pound of steam that corresponds to this velocity is

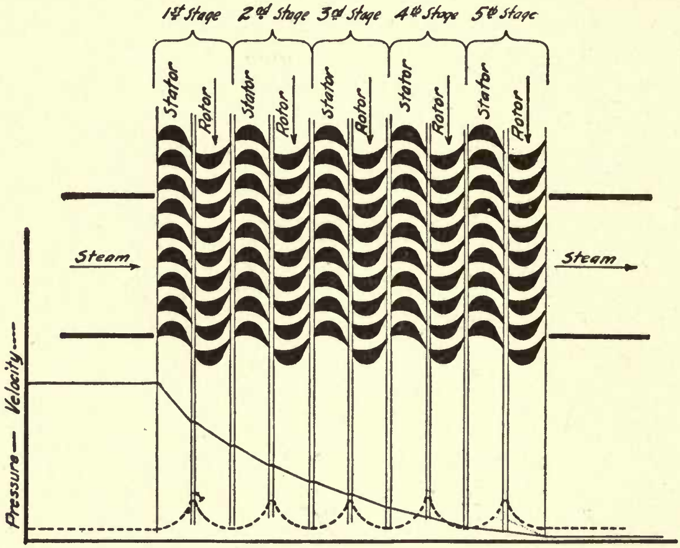
$$\frac{1}{2} \times \frac{1}{32} \times \overline{1884}^2 \times 1/778 = 71.3 \text{ B.t.u.}$$

With a pressure-drop from 165 to 1 pound absolute, in which there are available 323 B.t.u. for doing work, there will be $323/71.3 = 4$ pressure stages.

Comparing this with other types, it is seen that the number of pressure stages is the same as the number of velocity stages in the velocity-stage type. But as there are two rows of rotor buckets in each pressure stage, there will be twice as many rotor wheels as in the former type. With pressure-staging, there were 18 rows of rotor buckets, or more than twice as many as in the mixed type. This combination type is more efficient than the velocity-stage type, and at the same time it is more compact than the pure pressure-stage type. Due to these distinct advantages, the combination type is extensively used.

150. Multi-stage Reaction Type. — In the pure reaction turbine, all of the expansion of the steam occurs in the moving parts. At the present time, no pure reaction turbines are used. The so-called reaction turbine in use expands its steam both in the stationary and in the moving parts. It therefore employs a mixture of the impulse principle and the reaction principle. This mixed type is shown diagrammatically in Fig. 122. The steam enters from the left and passes through a set of stationary blades in which there occurs some drop in pressure. In passing through the next set of blades, which are moving, a further drop in pressure occurs. In this first set of rotor blades, the velocity generated in the stationary blades, and also that produced in the rotor blades is absorbed. Hence there is a drop in pressure

throughout the whole length of the turbine. As the velocity is used up as fast as it is produced, no very high steam velocity exists at any time during its passage through the turbine. The graphs show the drop in pressure and also the change in velocity as the steam passes through the turbine. The exact ratio of pressure-drop or velocity-change is not shown in the curves, as the scales used in the curves on the small cut are only illustrative.



Multi Stage Impure Reaction (Parsons)

FIG. 122

It has been shown previously (§ 143) that the velocity of the blades of a pure reaction turbine should be the same as that of the steam relative to the blades, in order to obtain the highest efficiency. This means that, for the conditions we assumed in § 147, the steam velocity to be used up per stage should be 471. This velocity corresponds to a kinetic energy per pound of steam of

$$\frac{1}{2}\times\frac{1}{32}\times 471^2 = 3470 \text{ foot-pounds.}$$

This is equivalent to 3470/778 = 4.46 B.t.u. per pound of steam, which is one-fourth of the value that was obtained for the im-

pulse turbine. For a heat drop of 175 B.t.u. (165 to 15 pounds absolute), it will require $175/4.46 = 39$ stages. Under the condensing conditions assumed in § 147, $323/4.46 = 72$ stages would be necessary. For the pressure-stage impulse type, the values were 10 and 18. Since the reaction turbine under discussion employs a combination of the impulse principle and the reaction principle, the values for the number of stages may be taken as a mean between the values for the pure impulse type and pure reaction type, that is 25 for the non-condensing condition, and 45 for the condensing condition.

The preceding computations show how much greater length the reaction turbine must have than the impulse turbine. The disadvantage of great length is offset to some extent by the fact that the loss due to friction is less in the reaction type, since the steam velocities are low.

151. Summary. — To recapitulate, assuming that the speed is 3000 r. p. m., and that the diameter of the rotor is 3 feet, the necessary stages for each of the various types is shown in the following table.

	Number of rows of buckets or blades on the rotor	
	Heat-drop of 175 B.t.u.	Heat-drop of 323 B.t.u.
Velocity-stage impulse	3	4
Pressure-stage impulse	10	18
Mixed pressure- and velocity-stage impulse	4	8
Pure reaction	40	72
Impure reaction	25	45

152. Change of Area of Steam Passage Space. — It is necessary to design a turbine so that the area of opening for the passage of steam gives the proper velocity at all times. As the pressure drops, the volume of steam increases. Hence there must be a very much larger area for the passage of steam in the later stages than in the first stages. This increase is accomplished in various ways. If the diameter of all the rotor wheels is kept the same, the area may be increased by having only partial peripheral admission of the steam in the early stages. That is, the nozzles or stationary vanes may extend only part way around the perim-

eter of the stator. In Fig. 123 *a* is the opening for the passage of steam in the first stage of the turbine. In the next stage, the opening extends farther around the stator, thus giving a large area for the passage of steam, and so on till, in the later stages, the opening extends entirely around the stator. The area may be increased by having full peripheral admission in all the stages, but having the length of nozzles and blades or buckets increase with each successive stage. Figure 124 shows this scheme. Or if full peripheral admission is given and the blades and nozzles are kept the same length, the area for the passage of steam may be increased by increasing the diameter of each successive stage, as shown in Fig. 125. In practice, combinations of these methods of increasing the area for the passage of steam are used. This increase in area to allow for the increase in the volume of steam passing, must not be confused with the increase in area in a

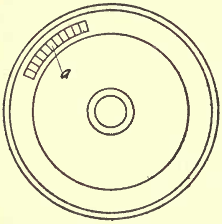


Fig. 123

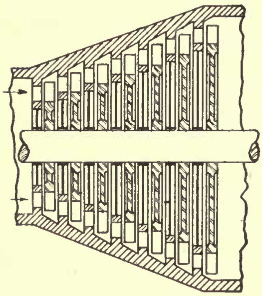


Fig. 124

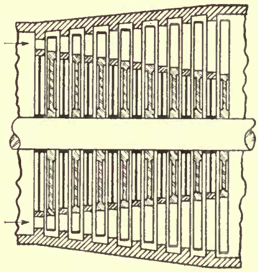


Fig. 125

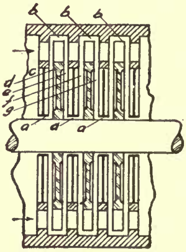


Fig. 126

velocity-stage turbine which is necessary to allow for the decrease in steam velocity for the different velocity stages.

153. Leakage. — The leakage of steam through an opening depends upon the difference in pressure that exists on the two sides of the opening, and upon the area of the opening. In most steam turbines, there are necessarily differences in pressure, and openings through which steam may escape. With the rotor of a turbine moving at a high speed, it is not possible to have tight joints at all places between the stationary and the moving parts, for then the friction between these parts would create an even

greater loss. In design and construction the clearances are kept as small as is consistent with economy and safety, but even then some leakage is sure to occur.

In Fig. 126, a difference in pressure exists between the two sides of the stator at *c* and *d*; hence there will be a leakage of steam at *a*. If there is a difference in pressure between *d* and *e*, a leakage occurs at *b*, and so on for the various stages. In the velocity-stage impulse turbine there is no difference in pressure after the steam leaves the nozzles; hence there is no tendency to leak at either *a* or *b*. In the pressure-stage impulse type, there is a difference in pressure between *c* and *d*, and therefore leakage at *a*; but there is none between *d* and *e*.

In the impure reaction type there is a difference in pressure between *c* and *d*, and also between *d* and *e*; hence there is leakage at both *a* and *b*. The leakage at *a* can be kept at a minimum by making the opening there as small as possible. This opening at *a* depends upon the closeness of the fit and upon the diameter of the shaft. Carbon or other form of packing is sometimes used here to give a close fit. In most reaction turbines, however, the shaft is very large, in fact it is even a drum, and in that case the area of the opening is quite large. Packing cannot well be used with large diameters.

In the reaction turbine the difference in pressure is not so great between *c* and *e* as in the impulse type. In the reaction turbine a leakage occurs at *b*; hence the clearance there is kept as small as is consistent with safety and economy. A certain amount of *spillage* occurs at *b* because the rotor throws some of the steam out by centrifugal force and it escapes through *g*, even if there is no difference in pressure between *d* and *e*.

154. Loss Due to Running at Partial Capacity. — Every turbine is designed to expand the steam from a certain initial pressure to a certain back pressure. This does not mean that the turbine will run only under the pressure conditions assumed in its design, but it does mean that the efficiency will be low if there is any great difference from the assumed conditions. For instance, if a turbine is designed to run non-condensing, and is operated condensing, it means that the efficiency will be considerably less than it would be if the turbine had been designed to run condensing. The reason for this is easy to understand.

Every nozzle and each steam passage is designed to carry a certain volume of steam with a certain velocity. We have shown that velocity is caused by drop in pressure, so that if the pressure-drops are not as designed, the velocities are not such that the efficiency will be a maximum.

When a turbine is designed for a certain load, and that load is greatly increased or decreased, it is evident that the efficiency will be decreased. In the design of the governor the aim is to make this drop in efficiency as small as possible, while at the same time maintaining uniform speed for all loads. It is also essential that the governor be reliable and therefore not too complicated. The simplest form of governor is a throttling governor, and many of the smaller turbines are equipped with that kind. However, with light loads, the throttling governor causes a large loss in efficiency because the range in pressure between admission and exhaust is so largely decreased.

On the larger machines some other means than throttling usually is used. If the machine is of the impulse type, it has stationary nozzles, and the governor can be arranged to open the proper number to take care of the load. If the machine had a single stage, or if a proportional number of nozzles could be kept open in the later stages of a multi-stage turbine, this would seem to be an ideal arrangement. But the large machines never are made single-stage and a governor to control all the nozzles in all the stages would be very complicated.

In practice, as in the ordinary Curtis turbine, the governor controls the nozzles of the first stage only. At light loads only a few nozzles of the first stage are open, while at maximum load all the first-stage nozzles are open. This arrangement assures good economy in the first stage at light loads, but as all the later-stage nozzles are open, there will be a loss there.

In the single-stage DeLaval turbine, a throttling governor is used, but an effort is made to maintain the best economy by having some of the nozzles controlled by hand-operated valves. By this method, most of the nozzles can be shut off at small load and opened up by hand for heavy load. The governor is incapable of controlling the load entirely if manual control is attempted.

In a reaction type of machine in which there are no stationary nozzles, the preceding method of governor-control cannot be used.

The Westinghouse Company, on their Parsons type of turbine, has attempted to secure good efficiency at light loads by having the governor admit the steam in puffs, which is the plan introduced by Parsons. That is, the steam is admitted at full pressure for a short time and then entirely cut off. The interval between puffs is practically constant, but the length of time the full steam pressure is on during the puff is controlled by the governor. In the later stages, the effect of this kind of governor approximates that of a plain throttling governor. In other makes of reaction turbine a throttling governor is commonly used.

Large overloads are often carried in the various types of turbines by turning full steam pressure into a later stage. In this way the machine is able to carry a large excess of load at a reduced efficiency. If an electric generator is attached to the turbine, care must be exercised that it is not overloaded long enough to get too hot. This method is used only in emergencies, and then only for short periods of time.

155. Summary of Losses in the Steam Turbine. — The losses that occur in a turbine have been mentioned in §§ 140, 147, 153, 154, and elsewhere. We shall now make a summary of the more serious causes of loss.

Friction Losses. Losses due to friction occur as follows:

(1) Between the shaft and the bearings, and in the packing rings where the turbine is made steam-tight. With proper design and construction this loss is quite small.

(2) Between the steam and parts of the turbine. The steam friction varies directly as the steam pressure, and as the square of the velocity between the steam and the parts. It also varies as the amount of surface in contact with the steam. Hence the amount of surface in contact with the steam should be kept as small as possible and the velocity should be kept as low as possible. The steam-friction loss in the first stages may be partially reclaimed in the later stages since the heat generated tends to raise the temperature of the steam. The steam friction occurs as the steam flows through the nozzles and blades.

(3) There is also a friction loss between the rotor discs and the steam surrounding them, at *d*, *e*, *f*, and *g* in Fig. 126. This loss is called ***windage;*** it may be reduced by having the rotor discs smooth and polished.

Leakage Losses. Leakage occurs wherever a difference in pressure exists on the two sides of an opening. It is thus seen that there may be a leakage out of the casing or through the joints between the pressure stages, or around the balance pistons of the reaction turbine (§ 160).

Another loss occurs under condensing conditions because air leaks into the low-pressure parts of the turbine. This air tends to lower the vacuum in the condenser, and may thereby cause a loss.

Disturbance of Flow. A breaking up of the jet of steam, and the consequent formation of eddies, causes a considerable loss in some types of turbines. This is one of the causes of the low efficiency of the velocity-stage impulse turbine (§ 147).

Lack of Proper Velocity. It has been shown that the proper relation must exist between the velocity of the buckets and that of the jet to obtain the highest efficiency (§ 143). If the velocity of the blades or buckets is not correct, there will be a loss. Throttling of steam by the governor will cause a loss. (See § 154.)

Exit Velocity. The turbine derives its energy by absorbing the steam velocity. A high velocity of steam in the exhaust entails a decided loss. The turbine should be designed and operated to extract nearly all the steam velocity, and to leave only enough in the exhaust to cause the steam to flow to the condenser.

156. Common Commercial Types. — A great many makes of turbines have been used in the past. Some of these were not economical and have been replaced. Others have ceased to exist for other reasons. At the present time there are a great many different forms in use, but space will not permit us to consider all of them. Some of the more common forms will be explained.

157. DeLaval Single-stage Steam Turbine. — The DeLaval single-stage turbine is the oldest of the types used at present. It dates back to the years 1880–1890. De Laval, the inventor of the cream separator, sought to drive that device by means of a direct-connected turbine. In his experiments he developed the type that bears his name. Some minor changes have been made, but its essential features remain the same. The DeLaval turbine used in America is manufactured by an American company that originally produced their machines under the DeLaval patents. The same company now makes a multi-stage machine, but it is not to be confused with the single-stage type.

The essential features of the DeLaval turbine are as follows:

(1) The expanding nozzle in which all the pressure-drop occurs.

(2) The rotor wheel which carries a single row of buckets.

(3) A slender, flexible shaft that carries the wheel and transmits the power to the gears.

(4) A set of reduction gears which lowers the speed so that it is usable.

One style of the DeLaval turbine is shown in Fig. 127. The

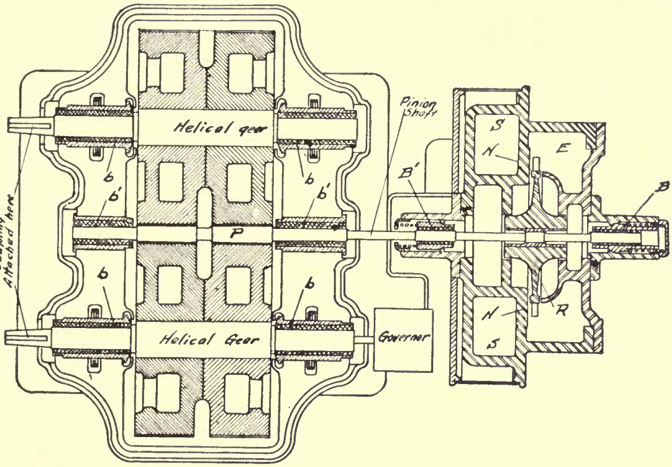


FIG. 127

bucket wheel R is mounted on a flexible pinion shaft which is supported by the bearings B, B'. This small shaft also carries the small pinion P which is supported by the bearings b', b'. The pinion meshes with the two large helical gears shown in the figure. The shafts that carry the large gears are supported by the bearings b, b. The shafts of the electric generators or pumps to be driven by the turbine are coupled to the large-gear shafts. The governor also is driven from one of these shafts.

The expanding nozzle is shown in Fig. 128. The steam enters from the left and the

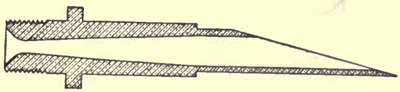


FIG. 128

area of the opening gradually increases to a size that produces the proper pressure and velocity. The amount of flare given to the nozzle is governed by the drop in pressure that is desired.

The nozzle used if the turbine exhausts to the atmosphere is called a ***non-condensing nozzle.*** If the turbine exhausts into a condenser, the nozzle is called a ***condensing nozzle.*** The flare in the non-condensing nozzle is less than that in the condensing nozzle.

One or more of the nozzles are open at all times; the rest are opened or closed by hand, depending upon the amount of the load. The governor is of the centrifugal type, and governs by throttling the steam. The nozzles are placed in the casing partition at N (Fig. 127). The steam chest is at S; the steam passes from it through the nozzles, then through the row of buckets on the rotor, and out into the exhaust space E. From E, the steam is led to the condenser or to the atmosphere.

We saw in § 145 that the bucket velocity of a single-stage turbine must be very high in order to extract a reasonable part of the kinetic energy of the steam. The high bucket-velocity causes large stresses in the rotor. The rotor disc therefore is made of high quality alloy steel and is very carefully designed and constructed. A sufficient factor of safety exists at the rated speed, but any large increase above this speed will cause failure in the wheel. It is customary to make the rotor weakest at a point just inside the rim where the buckets are attached, so that, if the breaking speed is reached, only the rim of the rotor will tear loose. When that happens, the speed will die down of itself because the buckets are gone. If the shaft becomes sprung, the safety bearings around the hub keep the main part of the rotor in place, so that no great damage results. The steel casing around the rotor is strong enough to keep any small fragments from breaking through; but if the whole rotor should break in two, it is likely that considerable damage would be done.

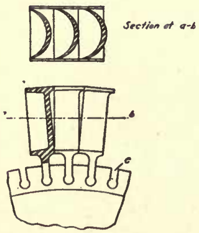


FIG. 129

The buckets are drop-forged. They are attached to the wheel as shown in Fig. 129. Slots are machined into the perimeter of the rotor and the buckets are forced into the slots from the side. Should an individual bucket become damaged, it may be removed and another put in its place.

At high speed, there is a tendency for the rotor to vibrate because it is impossible to get the center of gravity exactly in the center of rotation. There is a critical speed at which the vibration is a maximum. At a speed above this critical value,

the shaft or the bearings yield slightly, and the center of gravity of the rotor comes into the axis of rotation. The rotor then runs smoothly again. For smooth running, the speed must be either very much less, or considerably greater, than the critical value. Making the shaft small tends to reduce the critical speed. The shaft of this turbine is made very small so that it may give easily, and so run smoothly at its desired speed, which is normally above the critical speed. While the diameter of the shaft is small, it is ample to carry the power at the high velocity at which it runs.

The rotor speed in the smallest sizes is as high as 30,000 revolutions per minute. For a 300-horsepower turbine, the speed is 10,000 revolutions per minute. It will be seen that it is impracticable to run an electric generator or a pump at such speed, and to utilize the power developed, a speed reduction must be used. Helical gears are used for this reduction in order to insure smooth, quiet running. In some models, one large gear is used; in others there are two, as shown in Fig. 127. The latter necessitates two generators or pumps. When the gears are new, the loss of power in the reduction is small. This loss increases as the wear increases. Reduction gears are now used in some other turbines, though they were used originally only in the DeLaval turbine.

Aside from its use in the cream separator, the single-stage DeLaval turbine has been used in driving electric generators, centrifugal pumps and blowers. It is not used in sizes above 500 horsepower. The size is limited because the buckets would have to be unreasonably long, or else the diameter of the rotor would have to be too large, in order to get enough nozzles to play on the buckets.

158. The Multi-pressure-stage Impulse Turbine. — We have seen that there is a practical limit to the size of single-stage turbines. Because it gives the designer greater liberty of choice of speeds, velocities, and bucket lengths, the multi-stage impulse turbine is quite common in small and medium sizes. Moreover, the efficiency of this type is comparatively high in these sizes. For these reasons, it is quite common in sizes up to 5000 horsepower.

Since it has a number of pressure stages or *cells*, this kind of turbine is sometimes called the ***multi-cellular type.*** It is also

called the ***Rateau type,*** because Rateau was the first to develop it, but there is no essential difference between the Rateau turbine and the numerous other multi-cellular turbines.

Figure 130 shows an Economy turbine, made by the Kerr Turbine Company. As seen in the figure, there is a rotor shaft which carries a number of rotor wheels or discs. Each wheel runs in a pressure *cell*. The cells are separated by the heavy diaphragms shown at *a*. The joint between the diaphragm and the shaft is kept as nearly steam-tight as possible by making as close a fit as is practicable under the circumstances. The buckets are fastened

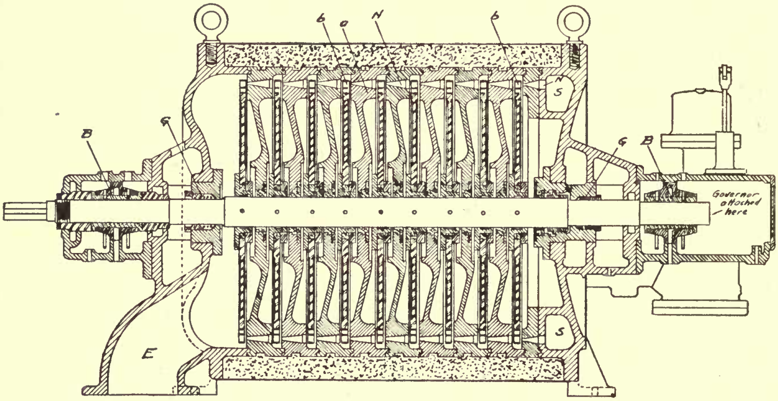


Fig. 130

on the rim of the wheels in much the same manner as in the single-stage DeLaval turbine. The shaft is supported by the bearings *B*, *B*. Nozzles are located in the cell partitions as at *N*. The number of nozzles increases from the first to the last stage, in order to allow for the increased volume of the steam as the pressure drops.

The steam enters the steam chest *S*, and passes through the nozzle *N*, where it is partially expanded. Leaving the nozzle, it passes through the first row of buckets *b*, on the rotor, then through the second set of nozzles, and so on till it arrives at the exhaust *E*. Leakage of steam from the high-pressure end of the turbine, and of air into the exhaust, is prevented by the stuffing-boxes *G*, *G*.

The casing is lagged with a non-conducting material to prevent the loss of heat by radiation. The bearings are equipped with

ring oilers. As in other impulse turbines, there is no difference between the pressure on the two sides of the rotor. Hence there is no tendency to leak steam over the ends of the buckets, and it is unnecessary to make the clearances between the buckets and the stationary elements small. This obviates the necessity of careful adjustments, and the danger of contact due to any unequal expansion of the parts. The diaphragms which compose the cell walls are made rigid so that they will not spring as a consequence of the differences in pressure on their opposite sides.

The turbine is supplied with a throttling governor of the centrifugal type, driven from a worm on the main shaft. In the smaller sizes it is direct-acting, that is, the position of the governor weights directly controls the opening of the throttle valve. In the larger sizes, it has been found desirable to have a *relay* arrangement by which the throttle valve is thrown by steam or by hydraulic pressure, which in turn is controlled by the position of the governor weights. This arrangement gives better speed regulation and does not require so large a governor. Considerable force is required to operate the large throttle valves.

Machines of this type made by other companies closely resemble the one just mentioned in essential features. Designers often increase the length of the blades or buckets with each successive stage, thus helping to allow for the increase in volume. The diameter of rotor wheels is sometimes made larger in the later stages for the same purpose. Various devices are used to keep the leakage from stage to stage at a minimum.

Carbon packing rings are sometimes used. In other makes, labyrinth joints are used. No large mechanical pressure is allowable at the contact points between stages. The result is that there is often a considerable loss from leakage of steam around the wheels, sometimes as high as 15 to 20 per cent.

Thrust bearings are also used to prevent end-play of the shaft, which might cause the buckets to rub and to become injured.

The practice of cutting holes in the rotor disc to insure an equalization of pressure on the two sides of the rotor is common. The shaft should be made large enough so that the speed at which it is to run is well below the critical speed, and then excessive vibration will not occur. It is necessary to use good workmanship and make the rotor well balanced; otherwise, severe strains would be induced at the high speeds at which the turbine is run.

It is not our purpose to describe the details of construction of all the various makes of turbines. It must be remembered that each make has its own minor variations in construction, but the preceding description holds in general for all makes of this type of machine.

159. The Curtis Steam Turbine. — The original patents for the Curtis turbine were issued about 1895, and the General Electric Company started production of this machine shortly afterwards. For several years it was built in this country only by them. More recently, however, the Curtis principle has been used largely in the high-pressure stages of some other machines. At first the General Electric Company built the larger turbines with the axis of the rotor vertical. The generator, which was direct-connected to the rotor shaft, was placed on top of the turbine, and the condenser was placed directly beneath it. This was claimed to be an ideal arrangement, and in some respects it was excellent, but all the weight of the rotating parts of both the generator and the turbine had to be supported by a step-bearing. With every precaution, and the best of design, this step-bearing would sometimes fail, and this failure often caused the buckets to strip. For very large, fast-speed turbines, it was difficult to secure sufficient mechanical rigidity for the bearing supports in vertical machines. The makers therefore have come to prefer the horizontal type. A great many of the vertical turbines are still in successful operation, however. The following description is taken from General Electric Co. Bulletin No. 4883.

> Figure 131 shows diagrammatically the progress of the steam in a Curtis turbine. Entering at *A* from the steam pipe, it passes into the steam chest *B*, and then through one or more open valves to the bowls *C*. The number of valves open depends upon the load, and their action is controlled by the governor. From the bowls *C*, the steam expands through diverging nozzles *D*, entering the first row of revolving buckets of the first stage at *E*, then passing through the stationary buckets *F*, which reverse its direction and redirect it against the second revolving row *G*.
>
> This constitutes the performance of the steam in one stage, or pressure chamber. Having entered the first row of buckets at *E* with relatively high velocity, it leaves the last row *G*

with a relatively low velocity, its energy between the limits of inlet and discharge pressure having been extracted in passing from C to H. It has, however, a large amount of unexpended energy, since the expansion from C to E has covered only a part of the available pressure-range. The expansion process is, therefore, repeated in a second stage.

The steam, having left the buckets G, and having had its velocity greatly reduced, reaches a second series of bowls H, opening upon a second series of nozzles I. Through these the

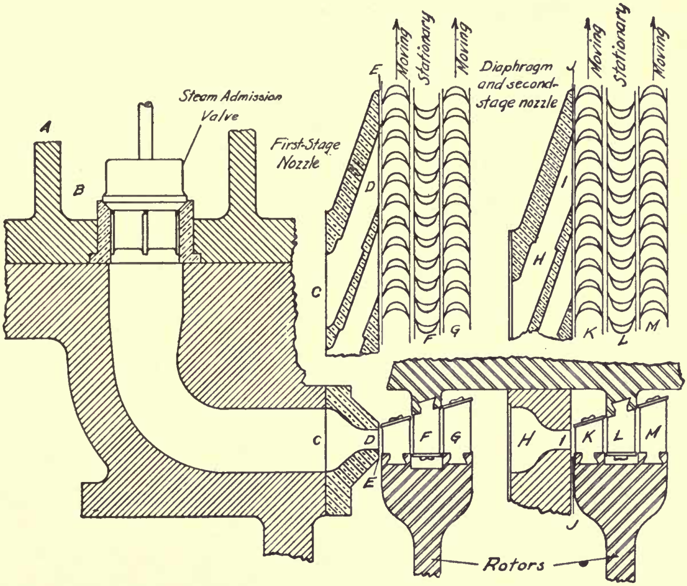


FIG. 131

steam expands again from the first-stage pressure to some lower pressure, again acquiring relatively high velocity in its expansion through these nozzles, leaving them at J and impinging upon and passing through the moving and stationary buckets K, L and M, precisely as in the first stage. Again the velocity acquired in the nozzle is expended in passing through the moving and stationary buckets, and the steam leaves the second row M with relatively low velocity.

This process is continued in most large turbines through several stages. Curtis machines as now constructed have a

single stage in the very small sizes, up to six or seven stages in the larger ones.

Again referring to Fig. 131, it should be noticed particularly that the pressure of the steam has not changed in its passage from *E* to *H*, that is the pressure is practically constant at all points in the stage. This fact leads to one of the principal structural advantages of the Curtis type. . . . For, since the pressure is uniform at all points, there is no tendency for the steam to pass elsewhere than where directed by the nozzles, *i.e.* through the buckets. Hence there is no necessity of maintaining a close clearance between the ends of the revolving buckets and the turbine casing. In practice free clearance is provided from one to two inches. The reaction type, to secure high economy, must be provided with a minimum clearance at this point.

Also since the pressure on both sides of each wheel, *i.e.* at *E* and *H*, Fig. 131, is the same, the wheel is in perfect equilibrium, there being no tendency for the steam to force the wheel in an axial direction. As this is true of each wheel, the entire rotor is in equilibrium, and there is practically no unbalanced thrust.

As the steam expands from stage to stage, its volume rapidly increases, and a greater area of steam passage must be provided. This is accomplished in two ways. First, by increasing the height of the buckets and second, by increasing the number and area of nozzles from stage to stage. Referring to Fig. 131, it should be noted that the primary admission nozzles *D* actually extend around a small portion of the first stage periphery; therefore only those buckets adjacent to the nozzles at any instant are carrying active steam. This applies equally to the stationary row and the second revolving row; in fact, the stationary or intermediate buckets, as built, extend over a small arc not much larger than the nozzle arc. In the second stage, however, the nozzle arc becomes longer and wider, thus permitting the flow of steam through a greater number of revolving buckets and necessitating a longer arc of stationary buckets. Finally, when the low-pressure stages are reached, the nozzles and stationary buckets extend all the way around the circumference.

As previously mentioned, greater area for the steam flow is

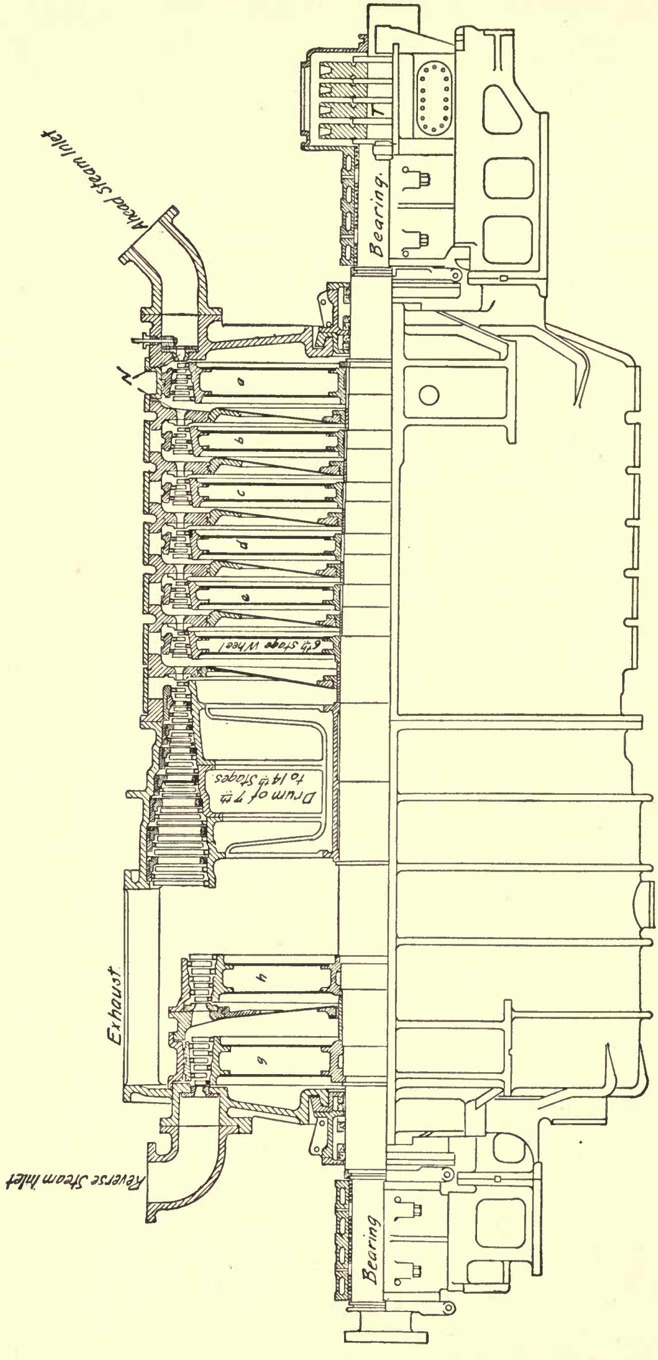
also provided by increasing the bucket lengths. For example, the first-stage, or high-pressure, buckets are generally less than an inch long, while those in the low-pressure stages may be eight or ten inches in length.

It will be noticed that the length of the second row of moving buckets in each stage, G and M, is greater than that of the first row. This is made so, not to allow for any expansion of the steam, but to provide for a decrease of velocity of the same volume of steam.

Customarily the buckets of the Curtis turbine are dovetailed into the rim of the rotor wheel, somewhat as shown in Fig. 131. At intervals the dovetail channel in the rim of the rotor is open for the insertion of buckets. These openings are afterwards filled with a spacing blank, and closed up. After the buckets are assembled a shroud ring is riveted to their outer ends. The function of this ring is partly to stiffen the complete row and to reduce vibration, but more especially to assist in retaining the steam flow in the bucket space. Centrifugal force tends to throw the steam out to the end of the bucket.

The governor is of the centrifugal type and controls the steam supply by opening and closing some of the nozzles of the first stage. Those nozzles that are open, are wide open, and those that are closed, are tight shut. This scheme is positive and reliable. It gives close speed regulation, and high efficiency at light loads. In addition to the governor, the machine is equipped with an emergency stop, whose function is to prevent excessive speeds, should the governor fail for any reason. It consists of an unequally weighted ring attached to, and revolving with, the shaft. At any speed up to the normal speed, the weights are held concentric with the shaft by springs, but at excessive speeds the force of the springs is overcome. Then the ring revolves eccentrically, and trips the valve mechanism, causing the main throttle valve to close instantly, thereby shutting off the steam supply.

Figure 132 shows a marine Curtis turbine. In marine service, the speed must be very much less than is common in land practice, if the rotor is coupled directly to the propeller shaft. To get this low speed, it is necessary to use very large diameters or a great number of stages. Usually both schemes are combined. As has been stated previously, the efficiency after the second stage of a velocity-staged impulse turbine is low, but in order to



Ahead Steam Inlet
N
Bearing
T
a
b
c
d
e
6th stage Wheel
Drum of 7th to 14th Stages
Exhaust
h
g
Reverse Steam Inlet
Bearing

FIG. 132

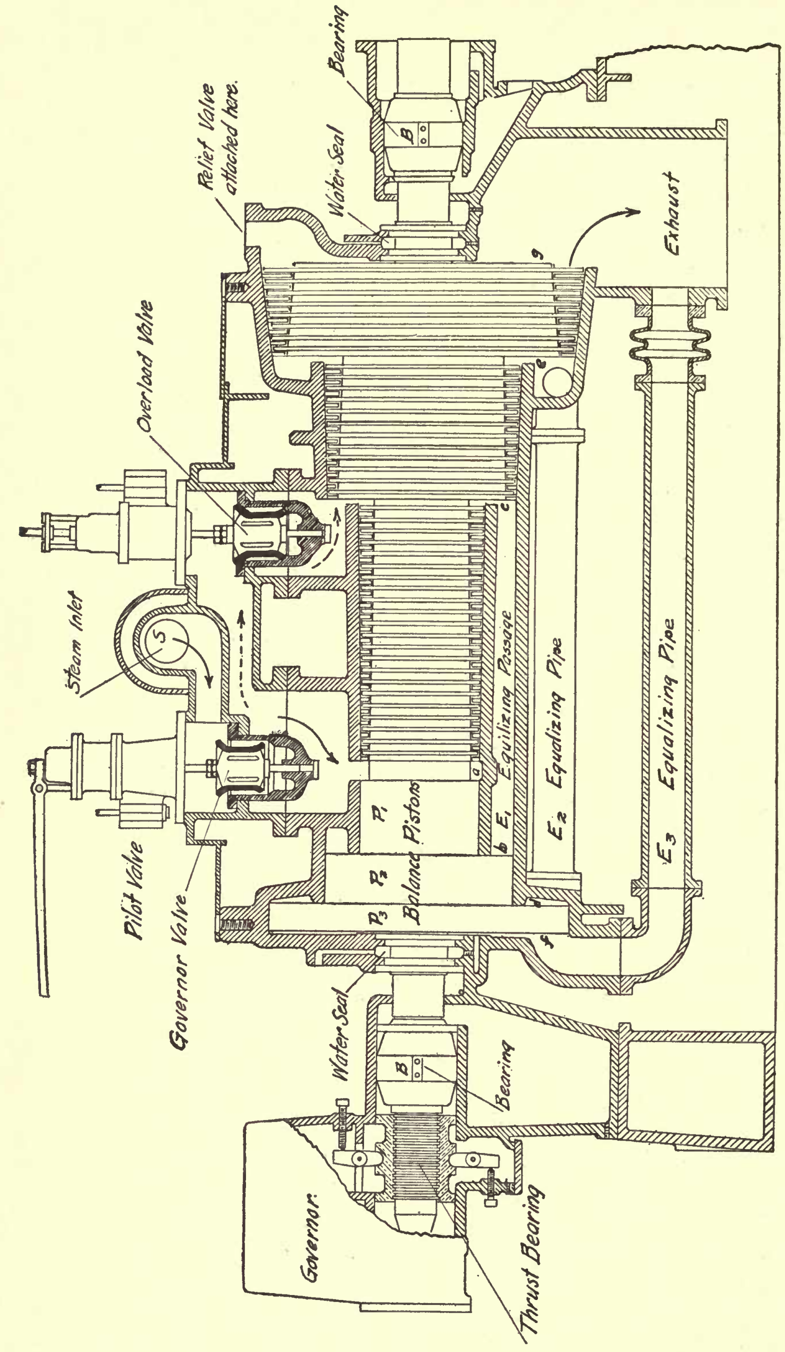
reduce the speed properly, it may be desirable to use more than two velocity stages, even if the efficiency is reduced. In land practice this concession is not often made. In the forward turbine shown in Fig. 132, there are four velocity stages in the first pressure stage, three velocity stages in each pressure stage from the second to the sixth, and two velocity stages in each pressure stage from the seventh to the fourteenth. In the reverse turbine, there are four velocity stages in each of the pressure stages.

It is customary in marine turbines to have the reverse turbine mounted on the same shaft with the forward or ahead turbine. It is put on the exhaust end of the shaft so that when it runs idle, the rotor is in a high vacuum and therefore offers as little resistance as possible to rotation. Since the friction loss varies as the steam pressure, the loss is not great for low pressures.

The efficiency during reverse operation is quite poor because there are only two pressure stages in the reverse turbine. The reverse is in use only briefly and rather unfrequently. Hence the the low efficiency of the reverse turbine is a negligible factor.

After a study of the previous types, Fig. 132 should be largely self-explanatory, hence no detailed description need be given. It is to be noted that the buckets of the seventh to fourteenth stages are carried by a drum. As the pressure on the right end of this drum is greater than that on the left end, it is seen that there will be an end-thrust of the shaft. This thrust is taken up by the thrust bearing *T*. The thrust bearing *T* is for the turbine only, and not for the propeller. On every propeller shaft there is a thrust bearing to take up the thrust of the propeller. The bearings are provided with a water jacket to prevent heating. In all large turbines, the bearings are cooled either by means of water or oil. When oil is used, it is often cooled by a device similar to the surface condenser.

160. The Parsons Steam Turbine. — The Parsons turbine is not only one of the oldest, but also one of the most common types. It is usually made in medium and large sizes. In small sizes the Parsons turbine is expensive, and is not very efficient. All of the previous types have operated on the impulse principle, but this one uses a mixture of impulse and reaction. It is ordinarily called a reaction turbine. There are no distinct nozzles, as in the impulse turbine. Instead, there are alternate fixed and mov-



Bearing
Relief Valve attached here.
Water Seal
B
Exhaust
Overload Valve
Steam Inlet
S
Pilot Valve
Governor Valve
Equilizing Passage
Equalizing Pipe
Equalizing Pipe
P_1
P_2
P_3
Balance Pistons
E_1
E_2
E_3
Water Seal
B
Bearing
Governor.
Thrust Bearing

FIG. 133

ing blades, and expansion occurs in both. It is made by the Westinghouse Company and by the Allis-Chalmers Company.

Figure 133 shows one style of Parsons Turbine. Instead of rotor wheels as in the previous types, the shaft carries a drum on which the moving blades are mounted. Steam enters at the steam inlet and passes through the governor valve to the left of the first set of blades. Passing through the alternate fixed and moving blades, it leaves through the exhaust outlet.

Full peripheral admission is used. The increase in passage area is accomplished by increasing the length of blades from stage to stage. The first-stage blades are very short, usually less than an inch in length. After a large number of stages, the blades are of considerable length. To avoid further increase in blade length, the diameter of the drum is increased, and the first blades on the larger diameter are made smaller so as to give the proper passage area. Progressing to the right the length of blades again increases, and again the diameter is increased. In the last stage the blades are quite long. In the larger machines, the final blades may be as much as a foot long.

Where the drum size is increased, there is an area exposed to steam pressure from the left. This pressure causes an end-thrust on the rotor to the right. To balance this end-thrust *dummy* or *balance* pistons are placed on the left end of the drum, each one being made of such a diameter that the steam pressure on its right side produces the proper force to the left. Initial steam pressure acts on the right of the smallest piston P_1. An equalizing passage, E_1, leads from the left of the second stage on the drum to the right side of the middle balance piston P_2, so that the same steam pressure exists at b as at c. In like manner, the pressure at e is the same as at d, on the right side of the largest balance piston P_3. A third equalizing pipe, E_3, connects the exhaust space with the left side of the piston P_3. Of course there is some leakage of steam by the balance pistons, but this is minimized by cutting annular grooves in the pistons and having rings on the casing extend into these grooves to form a labyrinth packing.

Since the pressure drops in both the fixed and movable blades, a leakage takes place, both between the ends of the fixed blades and the drum, and over the ends of the moving blades. It is therefore essential that the radial clearance be made as small as is safe. Even with the smallest clearance possible, there is bound

to be some leakage. The proportion of radial clearance space to the area of the steam passageway through the blades is proportionally greater with the short blades than with the longer ones. Hence more leakage occurs in the first stage than in the last. It follows that there is better economy in a reaction turbine in the low-pressure stages than in the high-pressure stages.

In some makes of Parsons turbines a shroud ring is fastened over the ends of the blades. In others, the shroud is left off, but the blades are lashed together by means of wires that pierce the blades. This wire is comma shaped in cross section, and the tail of the comma is caulked down on each side of the blades, thereby keeping them in the same relative position. These shroud rings or lashing wires do not add strength to resist the centrifugal force, but they keep down the vibration or *flutter* of individual blades. With long slender blades, the flutter might be more than the axial clearance and the contact at high speed might cause the blades to be torn loose. With only one blade loose, the whole system of blading might be almost instantly torn out. With this type of turbine, it is of the utmost importance that each blade be properly secured and adjusted. With the previous types, the stripping of blades may be localized to one stage, but in this, the damage is apt to be more general.

The machine represented in Fig. 133 is equipped with an overload valve. If the load is more than the turbine is ordinarily able to carry, this valve is opened, allowing high-pressure steam to enter at the second step, as shown by the dotted arrows. The steam consumption will be greatly increased, but a much larger load may be carried. Since the governor still controls, the speed may be considerably reduced. The overload valve is for emergency operation only, and is not supposed to be used often.

The two joints between the shaft and the casing are made tight by a water seal. As a turbine is ordinarily run condensing, there is a tendency for air to leak in at these joints. This leakage of air into the turbine is likely to produce a greater loss of efficiency than would a leakage of steam outward. The reason for this is that the vacuum in the condenser is greatly impaired by the air in the steam. If the seal is kept full of water, the leakage inward will be of the water, which will have little or no bad effect on the economy of the turbine. In some turbines low-pressure steam is used in place of water with the same effect.

As the weight of the rotor is very great, the bearings B, B must be well constructed and kept cool. As in the previous type, the bearings are usually kept cool by means of a circulation of oil. With small clearances, no end-play can be allowed. To keep the rotor from moving axially, a thrust bearing is used. The adjustment of the thrust bearing is made by means of the two screws shown in the figure.

The number of stages in a reaction turbine is very great, which necessitates a long rotor. With a long rotor, the changes in length due to temperature changes is considerable. This distortion increases with the amount of superheat. Hence little superheat can be used with some long reaction turbines.

To use more superheat, and also to limit the length of blades in the later stages, the designers sometimes resort to a scheme called ***compounding,*** *i.e.*, the turbine will be cut into two separate parts. The steam passes first through the high-pressure turbine, and then is led to the low-pressure turbine. If the high- and low-pressure turbines are both placed on the same shaft, the machine is called a ***tandem-compound turbine.*** Since they are on the same shaft, they must both have the same angular speed. Sometimes better results can be obtained by mounting each turbine on its own shaft and running the two at different speeds. With the latter arrangement, the machine is called a ***cross-compound turbine.*** In marine service, cross-compound turbines are sometimes used, and then each turbine is connected to its own propeller shaft.

In order to get rid of the balance pistons, Parsons turbines are sometimes made ***double-flow.*** In the double-flow turbine, the steam enters at the center of the casing and half flows to the right, while the other half flows to the left. The two halves of the drum are exact duplicates and any end-thrust on one half is balanced by the thrust on the other. While this adds to the total number of blades in the turbine, it does away with the dummy pistons. Figure 134 shows the double-flow arrangement, and is self-explanatory. Quite often the low-pressure turbine in the compound arrangement is made for double flow.

The governor used on the Parsons turbine made by the Westinghouse Company is of the ***blast*** type. As mentioned in §154, the steam is admitted in blasts or puffs. The speed of the governor is much less than that of the shaft, since it is reduced by

a worm gear from the main shaft. A diagram of this type of governor is shown in Fig. 135. The rod *C* is given an up and

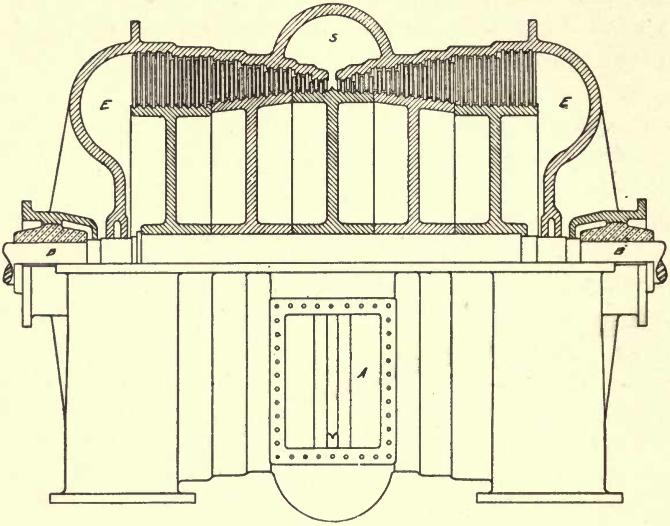


FIG. 134

down motion from an eccentric on the governor shaft. The pivots *D* and *E* being fixed, the reciprocating motion is communicated by means of the links and levers to the small pilot valve *A*. The recess on the valve *A* allows steam to enter periodically through the pipe, to pass through the ports, and to push up on the under side of the piston *B*. The fit around the rod running down from *B* is loose, so that the pressure soon drops. The vertical position of the point *F* is controlled by the position of the balls of the governor. This in turn controls the length of time the pilot valve admits steam under the piston *B*.

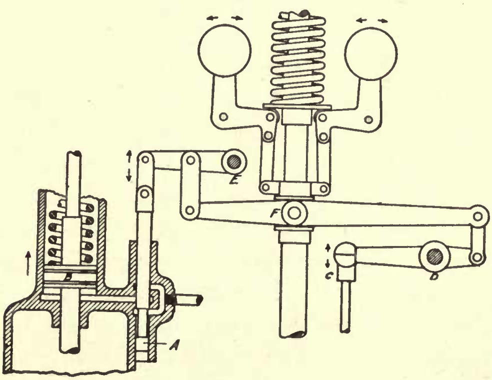


FIG. 135

The piston B is connected to the main steam valve of the turbine. When B is down, the main steam valve is wide open. When it is up, the steam is shut off. The length of the time it is up for each puff is seen to depend upon the position of the governor weights.

161. The Westinghouse Turbine. — Aside from the Parsons turbine just described, the Westinghouse Company has made for several years a turbine which they call the Westinghouse. The same type is made under other names, and has become quite popular abroad. It consists of one impulse stage, such as exists in the Curtis turbine, *i.e.*, one pressure stage, with two velocity stages, and the remainder of the turbine of the Parsons type. This Curtis stage may be used with a single-flow Parsons, or a double-flow Parsons, or with single-flow intermediate Parsons stages combined with double-flow Parsons in the final stages.

Figure 136 illustrates the Westinghouse type in which there is

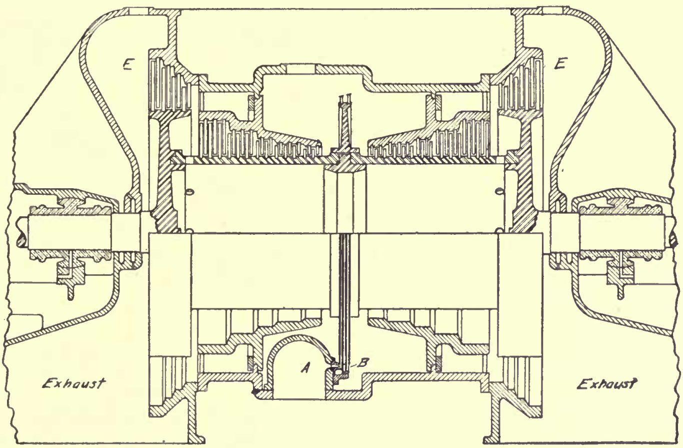


FIG. 136

a Curtis stage combined with a double-flow Parsons. The steam enters the inlet at A and passes through the first stage exactly as in the Curtis turbine; it then divides, half going to the right and half to the left, through the reaction stages, and comes out to the exhaust at the two ends of the casing. Partial peripheral admission is used in the Curtis stage and the governor controls the number of nozzles in use, as in the Curtis type.

There are certain advantages to be gained by the mixed Westinghouse type. First, the length of the rotor is shortened, because

the length taken up by the Curtis stage is only a small part of that which would be required by the same pressure drop in the reaction type. Second, the efficiency in the high-pressure part of the turbine is increased. We have seen that there is much leakage in the high-pressure stages of the Parsons type because the blade lengths are short in these stages, and the leakage is proportionately large. Third, it allows the governor to control the steam supply by shutting off nozzles, which is superior to either the throttling or blast governing.

Turbines are also in use that have one or two Curtis high-pressure stages, and low-pressure stages of the Rateau type. This combination is not common at present in this country, but it is found in some turbines in Europe.

There are many factors that determine the choice of type of a turbine. A satisfactory discussion is impossible here. Given the size, the kind of service, and the various operating conditions, the designer is able to say which type is the best.

162. Other Types. — In addition to the types heretofore described, there are various other small turbines in use. They are principally of the ***Pelton*** type, *i.e.* they are built on the same lines as a Pelton water wheel. Buckets are cut in the outer rim of a rotor and the steam enters them in a direction nearly tangential to the rotor. They are usually built in small sizes and are used for driving fans, blowers, and pumps. In such service simplicity counts for more than high efficiency.

163. Low-pressure Turbines. — The reciprocating steam engine utilizes economically a greater amount of the available energy of the steam at high pressures than at low pressures. That is, the efficiency of a reciprocating engine is relatively higher at a practical range of pressures above the atmosphere than below atmosphere. This does not mean that the non-condensing engine is more efficient than the condensing, but that of two engines, one taking steam at a high pressure and expanding to atmospheric pressure, and the other taking steam at atmospheric pressure and expanding down to that of a good vacuum, the former will be the more efficient. There is about the same amount of energy available for doing work in expanding from a medium boiler pressure down to that of the atmosphere, as there is in expanding from atmospheric pressure to that of a good vacuum. How-

ever, the amount of expansion is much larger in the second case. To utilize this great amount of expansion at low pressure, the cylinder would have to be very large; consequently a large cylinder loss would occur. With the turbine, the reverse is true, *i.e.* the turbine suffers relatively less loss at low pressures than at high pressures.

It has been shown by experience that the power out-put of a plant using non-condensing engines can be increased between 80 and 100 per cent by taking steam from the engines and sending it through what are called ***low-pressure turbines*** which exhaust into a notably excellent vacuum. This power increase is obtained from the steam without any extra coal cost or any increase in the size of the boiler plant, through changing the exhaust pressure from that of the atmosphere to that of the vacuum by the introduction of the turbine and its condenser. It sometimes happens that an existing plant, equipped with reciprocating engines, is to be enlarged, and in some instances the low-pressure turbine has been used to solve this problem. In case the existing engines are already condensing, it has been found that a net gain in power of 25 to 40 per cent may result by using their exhaust for the turbines, on account of the more perfect vacuum used with turbines since they are not subject to cylinder condensation and since the air leakage is less. The low-pressure turbine may take steam from engines, pumps, air compressors, hoists, etc.

164. Mixed-pressure Turbines. — The mixed-pressure turbine is much the same as the low-pressure turbine. It uses low-pressure steam, but it may also use some high-pressure steam at the same time. The high-pressure steam is admitted in varying amounts, to make up any deficiency in the supply of the low-pressure steam. This may be done by throttling the high-pressure steam down to the low-pressure before using it, or the turbine may be equipped with high-pressure stages to use the steam without throttling. The latter is the more efficient way of using the high-pressure steam.

165. Bleeder Turbines. — In some plants equipped with turbines, a supply of low-pressure steam is required for heating or for some manufacturing process. Rather than take high-pressure steam and throttle it down to the required pressure, the low-pressure steam may be drawn from the turbine at an intermediate stage. Then the machine is said to be a ***bleeder turbine.***

166. The Use of Superheated Steam. — Turbines are, in general, well adapted to the use of superheated steam. Comparative tests show a marked increase of efficiency when superheated steam is used. It has been claimed that this is due partly to a lessening of friction between the steam and the parts, but it is due mainly to the increased available energy in the steam which enters the turbine. Superheated steam also gives an increased efficiency when used in the reciprocating engine, but its use there is accompanied by an added difficulty in lubrication. With the turbine, lubrication is no more difficult with superheated steam than with saturated steam, since the steam-wet metal moving parts are not in rubbing contact with other metal parts. Superheating the steam used for turbines reduces the wear on the blading, compared with steam which is saturated or wet at the inlet.

167. The Marine Turbine. — The turbine has been used in marine service since the early years of its development. Many factors make the turbine particularly well adapted to this service. It occupies less space than a reciprocating engine, and it is lighter. It gives a uniform torque on the propeller shaft and does not cause as much vibration of the ship's hull as does the reciprocating engine. On the other hand, the turbine is a high-speed machine, while for good efficiency the propeller must be run at rather a low speed. Direct connection of the turbine to the propeller shaft is, of course, the simplest arrangement. When this is done, it is necessary to design the turbine to run as slowly as possible, and to design the propeller to run as fast as possible. Even then a compromise often has to be made: the propeller has to run too fast, and the turbine too slow, for best efficiency. Hence direct-connected turbines are limited to swift boats. One way out of this difficulty is to reduce the speed by means of gears. While this is sometimes done, it is not entirely satisfactory, since the gears are not highly efficient when worn.

A method of drive that is being used to a considerable extent is the ***electric drive.*** With this, turbines similar to those used on land are used to drive electric generators, and the current is used by motors direct-connected to the propeller shafts. This allows both the turbine and the propeller to run at the proper speed. It also permits great flexibility of arrangement, and convenience in steering and in maneuvering.

CHAPTER XI

GAS ENGINES

168. Introduction. — The small-sized internal-combustion engine is perhaps better known to the average person than any other prime mover. During the past twenty years it has exerted a very marked effect upon our manner of living. It has made possible the automobile, the motor truck, the gasoline tractor, and the airplane. It has replaced the more expensive, small hand machines and horse machines on our farms. It is indeed hard to estimate the value to mankind of the gasoline engine.

In plants in which there are combustible waste gases, such as those from the blast furnaces and coke ovens, large-size gas engine units have come into use, and they are there more economical than steam engines. In marine service, where space is a prime consideration, as in the submarine, the internal-combustion engine is generally used.

169. History. — Many years ago, men of an inventive turn of mind dreamed of gunpowder engines or explosive engines, and many patents were taken out for these devices. Records show that as early as 1680 Huyghens produced a working model of a gunpowder engine. It was of no practical importance. Many other inventors produced various forms of engines, but not till 1860 was an internal-combustion engine produced commercially. At that time Lenoir started building gas engines. In the course of a few years four or five hundred of these engines were built, but the engine was not very efficient as it lacked compression for the unignited gases.

The first really scientific work done on the gas engine was that of the French engineer, Beau de Rochas, who laid down the following four conditions as being essential to high efficiency.

(1) The largest cylinder volume, with smallest exposed surface, *i.e.*, the proper relation of diameter to length of stroke.

(2) The greatest possible rapidity of explosion, *i.e.*, the maximum piston speed.

(3) Highest possible pressure at the beginning of the expansion.

(4) The greatest possible expansion of burnt gases.

The same engineer proposed to obtain the above results by means of a single cylinder and to operate his engine upon the following cycle.

(1) Draw in a charge of mixed air and gas through an entire stroke.

(2) Compress this mixture during the next stroke.

(3) Ignite the compressed combustible mixture at the beginning of the third stroke, and expand the products of combustion during the stroke.

(4) Discharge the burned gases on the following stroke.

The above is known as the ***four-stroke cycle,*** or the ***Otto cycle,*** and is the one most commonly used. It will be considered more in detail later.

A few years after Beau de Rochas secured his patent, two inventors, OTTO and LANGEN, produced an engine that had a vertical cylinder with a free piston. The explosion of uncompressed gases drove this piston upward. On its downward stroke, a rack attached to the piston engaged, through a clutch, with a spur gear that drove the machinery. While this engine was more efficient than any produced before, it was noisy. Although several thousand were produced, the design was abandoned after a new design using compression of the admitted gases was produced by Otto.

The new Otto engine was shown at the Paris Exhibition in 1878. It operated on the cycle formulated by Beau de Rochas, and may be considered as the first modern gas engine.

A few years later, DUGALD CLERK brought out an engine with a ***two-stroke cycle.*** Many machines of this type are in use, especially in motor boats and the like.

Mention should be made of the Brayton engine, which was produced at about the same time as the Otto-Langen engine. GEORGE B. BRAYTON was an American, and many of his engines were used in this country. The principle of its operation is quite different from that of the others mentioned, but it need not be explained here.

The Diesel engine dates back to 1892, but it was not perfected until some time later. Since then, the semi-Diesel and other oil-burning engines have come into use.

These historical notes are given, not to explain the principles of operation of the early engines, but to indicate the length of the period in which the internal-combustion engine was evolved.

170. Cycles of Operation. — Modern internal-combustion engines operate upon either one of two cycles: the ***four-stroke***

cycle or the ***two-stroke cycle.*** These terms are usually abbreviated to *four-cycle* and *two-cycle.* The four-stroke cycle is sometimes called the Otto cycle, and the two-stroke cycle is occasionally known as the Clerk cycle. Each of these cycles is used with gas, gasoline, Diesel, semi-Diesel, and other oil engines.

171. The Four-stroke Cycle. — In the four-stroke cycle there are four strokes of the piston, two forward, and two backward. These occur as the shaft makes two complete revolutions. Figure

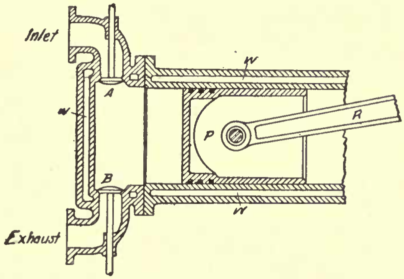


FIG. 137

137 represents an engine cylinder with its piston P. The inlet valve is at A, and the exhaust valve at B. When the piston is at its crank-end dead-center position, the volume to the left of it is the piston displacement plus the clearance. Both valves, A and B, are closed, and the space to the left of the piston is filled with a mixture of air and fuel.

During the first stroke, the piston moves from its crank-end to its head-end dead center, compressing the mixture into the clearance space. Near the head-end dead center, the compressed mixture is ignited, whereupon it burns, and its pressure suddenly increases.

On the second stroke the piston moves from the head-end dead center to the crank-end dead center, while the burnt gases expand and do work upon the piston. Near the end of the second stroke, the exhaust valve B opens.

As the piston moves to the left on the third stroke, the burnt gas is forced out to the exhaust. The burnt gas that is left in the clearance space is not expelled. At the end of the third stroke the exhaust valve closes.

At the beginning of the fourth stroke, the inlet valve A opens. As the piston moves to the right, a fresh charge of air and fuel

is sucked into the cylinder. At the end of the fourth stroke the inlet valve closes. This completes the cycle of operation.

Figure 138 shows the indicator card of the four-stroke cycle engine. Starting from E, at a little below the atmospheric pres-

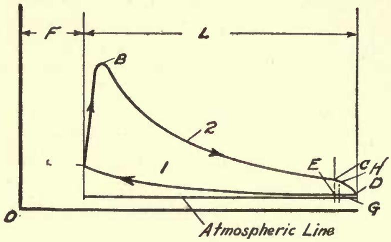


FIG. 138

sure, the charge is compressed during the first stroke to the pressure at the point A. From A to B, the charge is burned and the pressure rises. From B to C expansion takes place. From C to D the burnt gases are expelled. The suction stroke is represented by DE. The four strokes then represent *compression, burning and expansion, scavenging,* and *suction.* On the card, L represents the length of stroke, and F the clearance.

172. The Two-stroke Cycle. — This cycle is completed in one revolution. Figures 139 and 140 show a two-stroke cycle engine

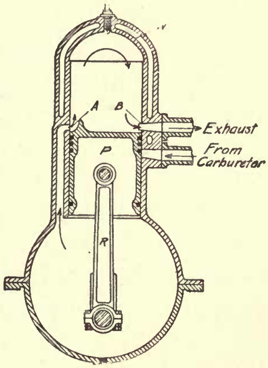


FIG. 139

such as is often used on motor boats. Due to the fact that the piston covers and uncovers the admission and the exhaust ports, it is often called a ***valveless engine.***

In Fig. 139, the piston is shown at the crank-end dead center. Both the inlet and exhaust ports are open. There is a small compression in the crank case and the combustible mixture of air and fuel is forced up and into the cylinder through the port *A*. There is a baffle on the top of the piston which deflects the incoming mixture to the top of the cylinder. At the same time the burnt gases from the previous stroke are escaping through the

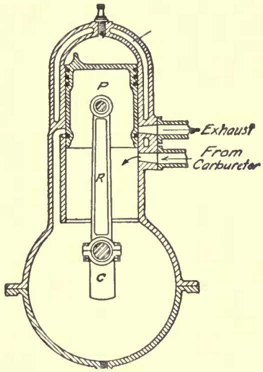


FIG. 140

exhaust port *B*. Naturally a little of the unburned gas will escape before the exhaust port is closed.

As the piston moves upward on its first stroke, it covers the ports *A* and *B*, and then compresses the mixture into the clearance space. At the head-end dead center, Fig. 140, ignition takes place, and the piston is forced downward on the second stroke by the pressure produced. The burnt gases expand until the exhaust port *B* is uncovered. They then escape to the exhaust. As the piston moves on down, the inlet port *A* is uncovered, and the fresh gas coming in at *A* sweeps out more of the burnt gas. As the piston moves upward a slight vacuum is formed in the crank case. When the piston gets nearly to the top of its travel, a port communicating to the carburetor or fuel and air supply is uncovered, and the combustible mixture is sucked into the crank case. Upon the piston's downward stroke this mixture is compressed enough to force it into the cylinder when the port *A* is uncovered.

In Fig. 141 another two-stroke cycle engine is shown. In the large size of these engines, separate pumps for air and gas are

used instead of the compression in the crank case. The engine is double-acting, and the exhaust port is placed around the cylinder midway between the two ends. The piston *P* uncovers this exhaust port near the end of each stroke. Gas and air is compressed in the pumps shown. The piston valves of the pumps deliver the gas and air alternately to the two ends of the cylinder.

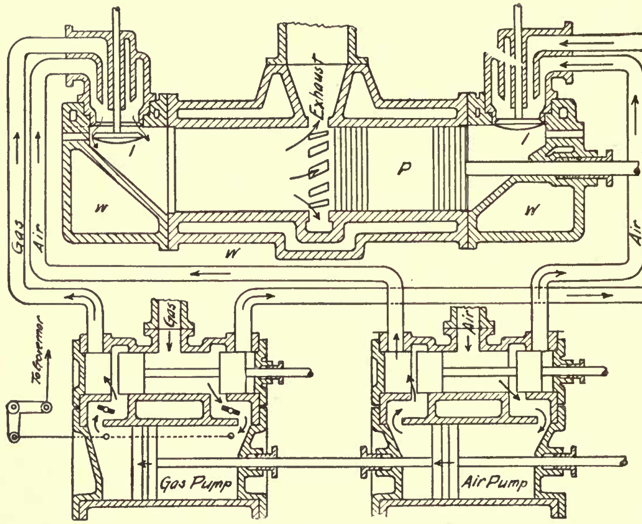


FIG. 141

The air and gas are mixed just before they enter the admission valve I. In Fig. 141, gas and air are compressed in the left ends of the pumps, and are forced into the left end of the engine cylinder. As the piston starts to the left, the left admission valve closes, and the mixture is compressed into the clearance space. At dead center the charge is fired, and expansion takes place as the piston moves to the right. When the piston uncovers the exhaust port, the burnt gases escape. In Fig. 141, the amount of gas forced into the cylinder is controlled by butterfly valves whose position is controlled by the governor.

The indicator card of the two-stroke cycle engine of Figs. 139 and 140 is shown in Fig. 142. Compression takes place from *E* to *A*, burning from *A* to *B*, and expansion from *B* to *C*. From *C* to *D*, and from *D* to *E*, the exhaust of the burnt gases takes place. The admission of the charge occurs from *H* to *D* and

from D to G. It is seen that the exhaust port opens a little sooner than the inlet port.

173. Classification from Fuel Used. — Internal-combustion engines are called by different names, according to the fuel used. We have the gas, gasoline, oil, Diesel and semi-Diesel engines. The fundamental principle is much the same in all of them, difference being largely a matter of detail in design and in the feed of the fuels.

In the earlier gas engines, city or coal gas was often used. Then, in the days of the natural gas booms in this country, gas engines became quite common. With these kinds of gas, it was

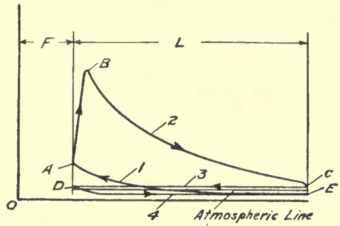


FIG. 142

not possible to give the mixture of fuel and air a very high compression, because the temperature during compression might be so high that ignition would occur before the end of the compression stroke. With the use of producer gas or blast-furnace gas the compression can safely be carried higher; hence we find it common to use a less amount of clearance with engines designed to use a lean gas for fuel. With a small clearance the compression is higher.

The internal combustion engine with which we are most familiar is the gasoline engine. Gasoline will vaporize partially at ordinary temperatures; hence a spray of the liquid fuel is mixed with the air as it goes to the cylinder. While this spray is commonly not all vaporized before reaching the cylinder, it is sufficiently vaporized so that an explosive mixture results, and the charge is fired. If the spray is fine enough and if sufficient time is given during the stroke, the gasoline will be very nearly all burned. Of course there are differences in gasolines, some kinds being more volatile than others. The device that introduces the spray into the

air intake is called a mixing valve, or a ***carburetor.*** With the less volatile liquid fuels, such as kerosene, heat is sometimes applied to help in vaporizing it before it is introduced into the cylinder.

When liquid fuels heavier than gasoline are used, it is common to spray them into the cylinder during the compression stroke. Often the spray strikes a hot plate in the cylinder and the fuel is sufficiently vaporized so that ignition can take place at the end of the stroke. The compression in low-pressure oil engines is no greater than in some gas engines, usually about 60 pounds per square inch.

In the Diesel engine the clearance is much less and a compression of 500 to 600 pounds per square inch is attained. Prac-

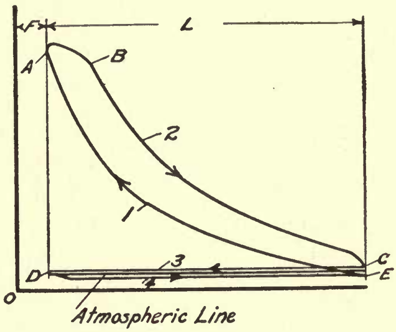


FIG. 143

tically all liquid fuels will partially burn at a temperature attained by the compression of air to 200 to 300 pounds per square inch. To prevent premature ignition, the liquid fuel is sprayed into the cylinder at the beginning of the forward or working stroke in Diesel engines.

Air alone is compressed during the compression stroke. At a pressure of 500 to 600 pounds its temperature will be in the neighborhood of 1000° F. This is sufficient to ignite and completely burn the oil that is sprayed in. The length of time the oil is injected into the cylinder usually is regulated by the governor. The Diesel engine may operate on either the four-stroke or on the two-stroke cycle.

The indicator diagram for the four-cycle Diesel engine is shown in Fig. 143. On the first stroke, air is compressed from a pressure a little below that of the atmosphere, shown at E, to 500 or 600

pounds per square inch, as shown at A. From A to B, the fuel is injected and burned. Expansion of the burnt gas takes place from B to C. The cylinder is scavenged from C to D, and a fresh charge of air is drawn in from D to E.

The high pressures necessary in the Diesel engine have been found troublesome from a mechanical standpoint, and there has been a tendency to reduce the high compression. With a compression around 200 to 300 pounds the term semi-Diesel is used. There is but little difference in the theory of operation between the Diesel and semi-Diesel. Oil is injected at the opening of the expansion stroke as before. However, on account of the

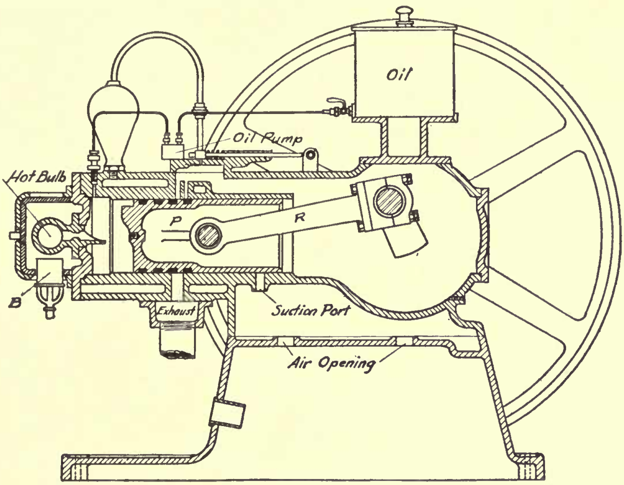


Fig. 144

lower temperature of compression, aid to the vaporization of the oil is given by the addition of a hot bulb located in the head of the cylinder.

Figure 144 shows this type of engine. Before starting, the bulb is heated by means of the burner B. After the engine has started the bulb will be hot enough without the aid of the burner. In Fig. 144, air is drawn into the crank case and compressed as the piston moves forward. With the piston near the crank-end dead center, the exhaust port is uncovered and the air is blown in through the ports at the top of the cylinder. As the piston returns to the left, the charge of fresh air is compressed. With

the piston on head-end dead center, the oil pump injects the fuel into the cylinder, and the spray strikes the lip of the hot bulb. The temperature of the bulb is high enough so that the fuel is ignited and practically all burned.

This engine operates on the two-stroke cycle. On the larger engines the four-stroke cycle is more common. On heavy loads, there is often a tendency to knock in the semi-Diesel engine. This is relieved by the injection of a small amount of water with the fuel, which does not seem to lessen the efficiency of the engine.

174. Efficiency. — The efficiency of an engine operating on the Carnot cycle is $(T_1 - T_2)/T_1$, where T_1 is the absolute temperature during combustion, and T_2 the absolute temperature of the gases in the exhaust. Of course none of our engines operate on the Carnot cycle, but their efficiency does depend upon the range of temperature in the cylinder. Other factors being the same, it is evident that the highest thermal efficiency will occur in that engine which has the largest range of temperatures during the working stroke. From this it is seen that those engines with the highest compressions, and therefore the highest combustion temperatures, will have the highest efficiency. This also explains the fact that gas engines using a lean gas, such as blast-furnace gas, may have a higher efficiency than those that operate on natural gas or gasoline vapor. Of all the internal-combustion engines, the Diesel has the highest thermal efficiency. The efficiency based on the brake horsepower ranges from 30 to 35 per cent. Other gas engines give somewhat lower efficiency.

175. Fuels. — Gas engines may operate on almost any combustible gas. Naturally the more expensive gases are but little used. Ordinary city gas is commonly a mixture of coal gas and water gas. As ordinarily produced it is too expensive for extensive use in gas engines. Where natural gas is plentiful and cheap, it is commonly used for gas engines.

At some blast-furnace plants, the gas from the furnace is used for fuel in the gas engines that drive the blowers. This gas is not of what we commonly call high quality, that is its heat content per cubic foot is much lower than that of natural gas or coal gas, but it gives excellent results when used in the engines.

The by-product gas from coke ovens is being used more and more as the efficiency of these plants is looked after. There are

also quite a number of plants, especially in the east, where producer gas is used for gas engines.

Natural gas, while it varies in composition, may be said to be composed mainly of marsh gas, CH_4. In some natural gases, there is a considerable free hydrogen and also an appreciable amount of olefiant gas, C_2H_4. The heat value of natural gas usually is between 900 and 1000 B.t.u. per cubic foot.

The illuminating gas used in cities varies widely in its composition, depending upon how it is made. It usually contains about 40 per cent H_2, 30 per cent CH_4, and varying amounts of CO and C_2H_4 besides CO_2 and N_2. The heating value averages from 500 to 600 B.t.u. per cubic foot.

Coke-oven gas contains about 50 per cent H_2 and 35 per cent CH_4. Its heating value is about the same as that of illuminating gas.

The principal combustible substance in blast furnace gas is CO, which ordinarily runs about 25 per cent. The heating value of blast-furnace gas is but little over 100 B.t.u. per cubic foot.

Producer gas, while it is variable, contains about 15 per cent H_2, and 25 per cent CO. It has a heat value of about 145 B.t.u. per cubic foot.

Practically all these gases contain a little O_2 and varying amounts of CO_2 and N_2. These substances in the gas add no heating value to it.

The liquid fuels used in internal-combustion engines vary from crude oil to more refined products, such as gasoline. Many Diesel engines seem able to burn crude oil very well, but some of the semi-Diesel and oil engines do better on the more volatile product, such as kerosene. The great demand for gasoline has led to a gradual lowering of the flash-point of this product. Carburetors that used to give good results with the gasoline sold a few years ago, now have trouble in using the commercial grades. The development of carburetors has had to keep step with the change in the quality of the product they have to handle.

176. The Gasoline Carburetor. — There are a great variety of mixing valves or carburetors on the market. We cannot hope to describe all of the various types in this course. Only one simple form will be described, but the fundamental principle is the same in all. This principle is to divide the liquid fuel into as fine a

spray as possible and to mix it thoroughly with air. Some of the liquid is evaporated, but it is doubtful whether it is ever all vaporized.

Evaporation is not necessary if the liquid particles are finely enough divided and are thoroughly mixed with the air. Even solid combustible matter is explosive when mixed with air, as is shown by flour-mill explosions and the explosions in mines due to dust of inflammable materials.

Figure 145 shows a gasoline carburetor. The main body of the carburetor *B* is partly filled with gasoline. The height of

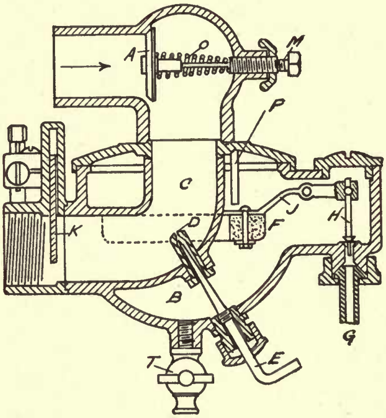


FIG. 145

the liquid is kept very near to a constant level by means of a float *F*, which is attached by means of a lever to the valve *H* which leads from the supply. Air enters through the opening at the top and passes down through the passage *C*. Turning to the left, it passes out to the pipe leading to the engine. A spray of liquid is injected into the air through the needle valve *D*. The opening in the needle valve *D* is adjusted by the handle *E*. The opening of the needle valve into the air passage is a little above the liquid level in *B*, so that no gasoline runs out unless there is a current of air to suck up the liquid. There is at all times an opening for the air to enter the carburetor at *A*, but this opening may be increased when a large supply is needed by the suction pulling back the valve *A*. The valve *A* is held on its seat by a light spring *O*. The tension in the spring *O* may be

adjusted by turning the screw M. By the proper adjustment of the air valve A and the needle valve D, any richness of mixture may be secured. The amount of mixture of air and gasoline leaving the carburetor to enter the engine is regulated by the throttle K. The bowl of the carburetor may be drained by opening the cock T.

177. The Gas Producer. — A large amount of publicity has been given to the subject of gas producers for the gas engine, and much research work has been done on the subject. It is only just, however, to say that the producer plant has been somewhat of a disappointment in America. While trial tests show up remarkably well, and the claims of makers are unusually good, actual experience has shown that the time has not yet come when they can replace the steam plant. It is not safe to predict as to the future, and every power plant engineer should be somewhat acquainted with the subject.

When air is passed through a bed of hot carbon, combustion takes place. If there is sufficient air, the combustion is complete and CO_2 is formed. If not enough air is supplied, the burning is only partial and CO is formed. This CO may later be burned to CO_2 by the addition of more air, which is done in the engine cylinder in the case of the producer and engine plant.

If steam is passed through a hot carbon bed, a decomposition of the steam takes place. The hydrogen is liberated as H_2 and the oxygen combines with the carbon to form CO. Both of these gases are valuable as fuel, and the mixture is often called ***water gas***.

When air is passed through the hot carbon bed, and CO is formed, heat is generated, so that the bed gets hotter and hotter. On the other hand, when steam is passed through, heat is absorbed and the bed gets cooler and cooler. By the proper proportioning of air and steam passed through, it is possible to keep the fuel bed at the proper temperature. This is what is done in the gas producer. It is evident that most of the gases in producer gas will be CO, H and N, the nitrogen coming from the air and being inert.

The fuel used in the producer may be coke or coal. Better results are obtained by using anthracite coal than by using bituminous coal. This is partly due to the fact that bituminous coal

tends to cake and needs constant working or poking to keep holes from burning through the cake, thereby letting excess air get through. Moreover, bituminous coal gives off various tars when it is heated. If these are not removed, they clog up the pipes and the engine. The removal of the tar is not easy. It is sometimes done by throwing the tarry material out of the gas by centrifugal force by means of a kind of fan arrangement, or by passing the gas through scrubbers. Devices have been tried

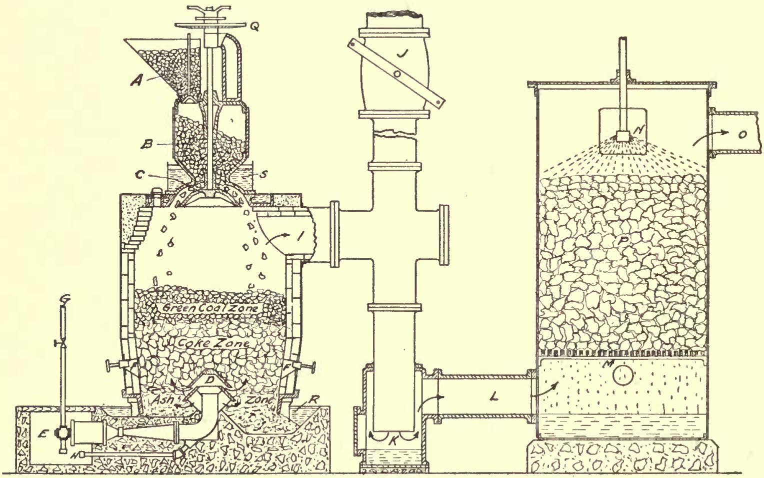


FIG. 146

whereby the distilled products are passed through the hot fuel bed and their composition thereby changed. This last method resembles the underfeed furnace used in steam plants.

Figure 146 shows a form of gas producer. Coal is fed into the hopper *A*. From this, it is dropped into the chamber *B*. Passing out of the bottom of *B*, it is scattered in a uniform layer over the fuel bed by means of the spiral spreader *C*, which is rotated by the bevel gear *Q*.

The fuel bed may be divided roughly into three zones. The top zone is the green-coal zone, where distillation takes place. The volatile products pass on out with the other gas.

As the volatile products are driven off, and the bed settles, the fuel reaches the coke zone. It is here that the burning and decomposition mentioned above take place. As the carbon is burnt

out, the ash settles to the bottom of the producer, where it is raked out through the water seal *R*.

Air from the duct *E* is led to the bottom of the fuel bed through *D*, and passes up through the coke zone and the green-coal zone. Steam is admitted to the air supply through the pipe *G*. Holes are provided in the furnace walls at *F* for the working of the fuel bed.

The gas leaves the producer through the opening at *I*, and passes down the pipe shown to *K*. From *K* it passes through the pipe *L* to the wet scrubber. The wet scrubber is filled with coke, *P*, which is continuously sprinkled with water from the nozzle *N*. The gas passing up through the wet coke is cooled and deposits dust, tar and other impurities. The gas leaves the wet scrubber at *O*, and may go either directly to the engine or else to a dry scrubber. The dry scrubber is filled with wood shavings or excelsior, which takes out the remaining tar.

Upon starting the producer, the gas is vented to the roof through the valve *J*. As soon as the quality of the gas becomes good enough, the valve *J* is closed and the engine is started. The coal in *B* is kept cool enough to prevent it from burning, by a water-jacket *S*. The producer is lined with firebrick.

Air either may be blown into the producer, or it may be drawn through by the suction of the engine. In the former case a storage tank for the gas is necessary. With the suction type the storage is unnecessary, as the engine draws through only what it needs. When a water seal is used at the bottom, the producer is called a wet bottom producer. The poking of the fuel bed may be done by hand or mechanically.

178. Cooling of Cylinders. — The cylinder walls of the internal-combustion engine must be kept cool enough to insure proper lubrication. This cooling is commonly done by circulating water through a jacket around the cylinder, as shown at *W*, in Figs. 137, 139, 140, 141 and 144. As far as the efficiency of the engine is concerned, the hotter the cylinder walls the better, so that it is evident that they should not be cooled any more than is necessary to insure lubrication. In small engines, the cylinder is sometimes placed in the lower part of a hopper filled with water. Fresh water is added as it boils away in the hopper.

Another method that is used occasionally to cool the cylinder

is by having the outer walls of the cylinder and head covered with fins. These present a large surface to the air and the heat is radiated from them. To increase this transfer of heat, a current of air is kept moving over the surface of the fins by means of a fan. This latter method is called air-cooling. Figure 147 shows an air-cooled cylinder.

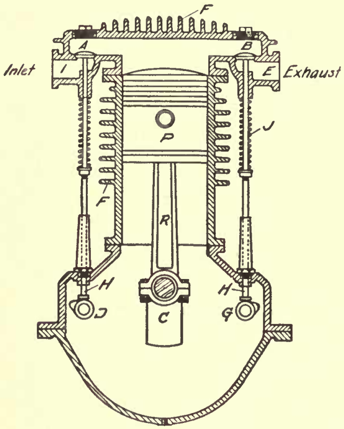


FIG. 147

179. Ignition. — The charge of combustible mixture in an internal-combustion engine is fired in various ways. A method formerly used quite extensively but now not very common is the ***hot-tube*** method. This method is illustrated by the sketch in Fig. 148. The tube which connects with the cylinder is heated by a gas jet. Arrangement is made so that the flame can be shifted along the tube, heating it closer or farther away from the cylinder.

To explain how the scheme works, suppose the charge has been fired. The tube will then be filled with burnt gas. After the fresh charge has been drawn in and compression started, the fresh gas is forced into the tube. When it is compressed into the tube far enough to strike the heated part, ignition occurs. Naturally, this scheme can be used only when the load is constant.

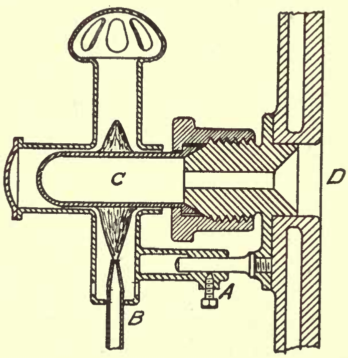


FIG. 148

As has been explained previously, ignition may be had by using a ***high compression,*** so that the temperature of compression will be high enough to fire the charge. This method is used mostly in oil engines and is assisted in those using the lower pressures by a hot bulb or plate. The hot bulb acts somewhat in the same manner as the hot tube just mentioned. With gas

or the more volatile liquid fuels this method is not satisfactory, since early ignition is apt to occur.

The most common method is ***electric ignition.*** There are two general types of electric ignitors, the ***jump-spark*** and the ***make-and-break.*** In the former, a spark plug is used (Fig. 149), which has two fixed terminals exposed to the gases of the cylinder. At the proper time for ignition a current with voltage high enough to jump the gap is introduced in the circuit. The heat of this spark ignites the charge. The details of timing and of producing the current will not be discussed here, except to say that the current may be furnished either by dry-cell or storage batteries, or by a magneto.

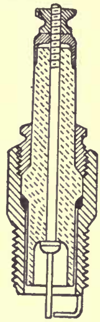


FIG. 149

In the make-and-break system, two ***electrodes*** are brought into contact within the cylinder and are separated at the proper time for ignition. As the circuit is broken a spark is formed between them. The details of the many schemes used will not be discussed here. This make-and-break system does not require as high an e.m.f. as the jump-spark system, but it is limited to the slower engine speeds.

180. Valves. — The earlier types of gas engines were equipped with slide-valves. These required lubrication, which is somewhat difficult at high temperatures. Except for a few engines that use ***sleeve-valves,*** gas engines of today have the so-called ***lifting*** or ***poppet valves,*** which are commonly lifted by cams on a cam shaft. In the four-stroke cycle engines the cam shaft is geared to run at half the speed of the crank shaft, so that the cam shaft makes one revolution per cycle. The cams are placed on the cam shaft so that the valves open and close at the proper time. Each valve is kept seated by means of a spring around the valve stem. Figure 147 shows the cams, and how they are made to lift the valves. In some of the slower speed engines, the inlet valve is not opened by means of a cam, but by the suction inside the cylinder. With this arrangement a strong spring cannot be used to seat the valve.

181. Governing. — As far as the governor itself is concerned, the gas-engine governor does not differ materially from the steam-engine governor. Both centrifugal and inertia governors are used.

Depending upon how the governor regulates the amount of work done in the engine cylinder, we have three general types of gas engine governors: (1) hit-and-miss governors, (2) quantity governors, and (3) quality governors.

Hit-and-miss Governor. With this type of governor there is a working stroke for every cycle under conditions of maximum load. At lighter loads the governor mechanism fails to admit a charge occasionally, giving what might be termed a blank cycle in which no work is done by the cylinder. This drops the speed of the engine and the governor acts so that fuel is again taken in as before. With this scheme the engine either operates under conditions of maximum efficiency, or it does not fire at all. This method of governing gives better economy at light loads than the other methods, but it does not give close speed regulation. When the engine *misses,* the exhaust valve is commonly held open so that there is no work done in useless compression.

Quantity Governor. For every engine there is a ratio of gas to air, which is nearly constant, with which the engine gives the best efficiency. With the quantity governor, this ratio is kept (theoretically) constant. Regulation is accomplished in two ways: (1) by the cut-off governor, and (2) by the throttling governor. With the cut-off or throttling governor, the normal mixture is allowed to enter the cylinder only during a part of the suction stroke at light loads. The length of time the mixture is admitted is controlled by the governor. With the throttling governor, the normal mixture is taken in during the whole of the suction stroke, but the opening is throttled so that not as much enters at light loads as at full load.

Quality Governor. This governor changes the ratio of the fuel to the air at different loads. At full load, a rich mixture is used, and at light load a lean mixture. Mechanically, this scheme is quite simple, but it has the disadvantage of giving low efficiency at light loads. If the mixture gets too rich or too lean, it may be impossible to secure ignition. Oil engine governors commonly control the amount of oil admitted per working stroke. This is seen to be the same as the quality-governing scheme.

It should be mentioned that the speed of an engine can be regulated by changing the time of ignition. With either too early or too late ignition the full power is not developed in the

cylinder. The speed of motor-boat engines is often controlled in this way. With high speeds, the spark should occur earlier than in low-speed engines. With a variable-speed engine, such as exists in an automobile or truck, the time of ignition should be adjusted to the speed in order to get the best results.

182. Determination of Horsepower. — The indicated horsepower of a gas engine is determined in the same way as for the steam engine, with the exception that for four-stroke cycle engines only half the r.p.m. is used in the computation. If a hit-and-miss governor is used on the engine, the number of hits per minute must be counted rather than the r.p.m. of the shaft.

183. Multi-cylinder Engines. — The single cylinder, single-acting four-stroke-cycle gas engine has one impulse stroke in two revolutions. The double-acting steam engine has two impulse strokes per revolution. Thus it is seen that the single cylinder gas engine has a much greater variation in angular acceleration of crank shaft than does the steam engine. For some kinds of service this variation in angular velocity is immaterial; in other cases it is a serious disadvantage. For instance, in the generation of electric current to be used for lighting, a single-cylinder engine is impracticable, unless an exceedingly heavy flywheel is used. To approximate the uniformity of torque that exists in a single-cylinder steam engine, it is necessary to use four cylinders on the gas engine.

In automotive service, a fairly uniform torque is desirable, and therefore four or more cylinders are used. If more than four cylinders are used, there will be less variation in the angular torque, and the engine speed may be controlled more easily by the throttle. Too large a number of cylinders may cause a decrease in the efficiency of the engine. This may be explained by the fact that with a multi-cylinder engine, there is more area of cylinder wall exposed to the burned gas for the same volume of gas than there is in the single-cylinder engine. Principle I of §169, as set forth by Beau de Rochas, is a statement of this same fact. While a lowered efficiency may result from the use of a large number of cylinders, this loss may be more than compensated by the added smoothness of running. Many of the higher grade automobiles made at the present time are equipped with six, eight, or even twelve cylinders.

A REASON WHY

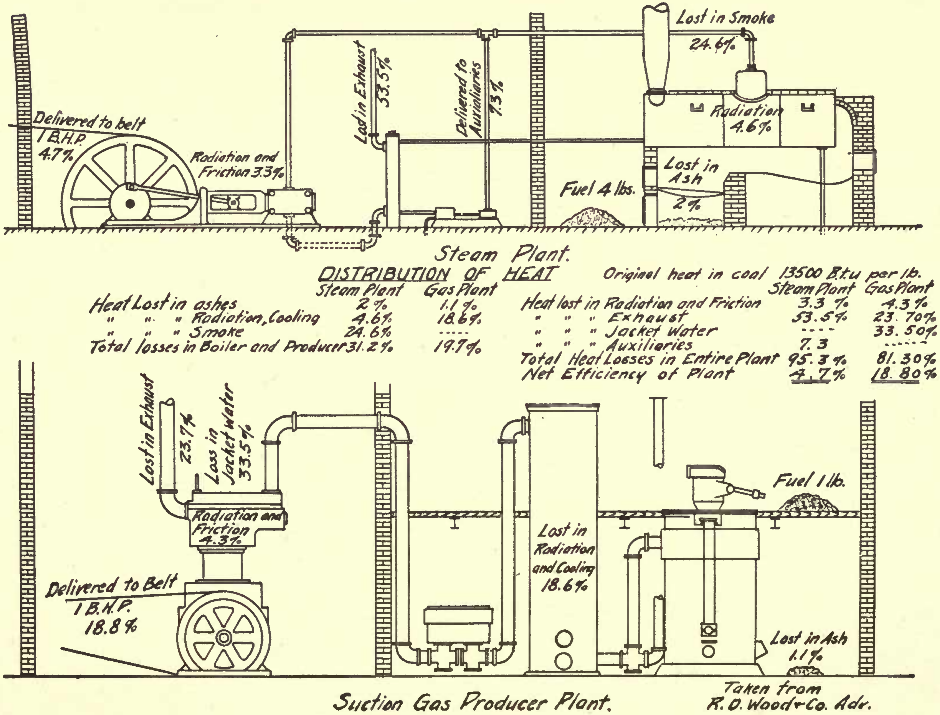


FIG. 150

PROBLEMS

1. (*a*) What is the pressure in pounds per square inch that corresponds to a mercury column 16 inches high?

(*b*) What is the atmospheric pressure when the barometer reads 27.4 inches?

2. A steam gage is used to show the pressure in a steam line and is attached as shown in Fig. A. If the small pipe leading to the gage is full of water and the gage reads 183 pounds, what is the pressure in the steam line?

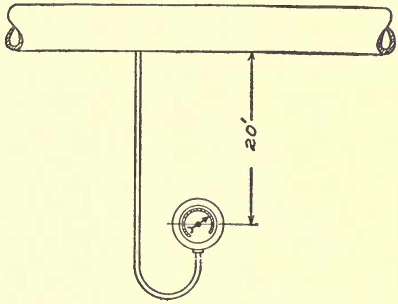


FIG. A

3. If the pressure gage on a boiler reads 150 pounds and the barometer reads 29.3 inches, what is the absolute pressure in the boiler in pounds per square inch?

4. The vacuum gage on a condenser reads 27.2 inches and at the same time the barometer reads 29.1 inches.

(*a*) What is the absolute pressure in the condenser in pounds per square inch? (*b*) What is the vacuum-gage reading reduced to a 30-inch basis?

5. A condenser with its air pump is guaranteed by the manufacturer to produce a vacuum of 28.5 inches (on the basis of a 30-inch barometer). During the acceptance test the barometer read 28.73 inches. What should be the vacuum-gage reading to maintain the guaranteed vacuum?

6. A boiler-feed pump is located 14 feet below the water line of the boiler. The pump draws water from a tank located 7 feet below the pump cylinder. If the pressure in the boiler is 40 pounds gage, neglecting friction losses due to the flow of water, etc., what is the least total head the pump must act against: (*a*) In feet? (*b*) In pounds per square inch? (*c*) What is the least height of water level in a standpipe above the boiler in order that the water will flow into the boiler by gravity?

7. Reduce: (*a*) A temperature reading of 50° Centigrade to the corresponding Fahrenheit reading. (*b*) A temperature of 320° Fahrenheit to Centigrade. (*c*) 75 (great) calories to B.t.u. (*d*) 46 B. t. u. to calories. (*e*) 33000 foot-pounds to B.t.u.

8. If 1,576,000 B.t.u. are given to an engine in an hour and if the engine can convert 6 per cent of this heat into work, what is the horsepower of the engine? (One horsepower is 33,000 foot-pounds per minute.)

9. A sample of Indiana coal gave the following proximate analysis:

Moisture = 3.81%,	Fixed carbon = 76.16%,
Volatile combustible = 13.62%,	Ash = 6.41%.

The same sample when dried gave the following ultimate analysis:

Carbon = 84.26%,	Oxygen = 1.73%,	Sulphur = 1.22%,
Hydrogen = 4.38%,	Nitrogen = 1.75%,	Ash = 6.66%.

The oxygen calorimeter gave a calorific value of 14682 B.t.u. per pound of dry fuel. Find the calorific value of the preceding sample per pound of *dry* fuel, from (*a*) the proximate analysis, (*b*) the ultimate analysis.

10. A sample of West Virginia coal gave the following proximate analysis:

Moisture = 4.85%, Fixed carbon = 68.36%,
Volatile combustible = 16.31%, Ash = 10.84%.

The same sample when dried gave the following ultimate analysis:

Carbon = 80.34%,	Oxygen = 3.11%,	Sulphur = .49%,
Hydrogen = 4.00%,	Nitrogen = 1.05%	Ash = 11.01%.

The oxygen calorimeter gave for a similar dried sample a calorific value of 14,180 B.t.u. per pound. Find the calorific value per pound of *dry* coal from

(*a*) the proximate analysis (*b*) The ultimate analysis.

11. Find the theoretical weight of air required to burn completely a pound of the dry coal of Problem 10.

12. A boiler test was run using the coal from which the sample of Problem 10 was taken. During the test the analysis of dry flue-gas was as follows:

Carbon dioxide = 8.58%,	Carbon monoxide = .05%,
Oxygen = 11.32%,	Nitrogen = 80.05%.

Find the approximate weight of air used to burn one pound of dry coal.

13. In the test of Problem 12, the temperature of air entering the furnace was 64° F., and the stack temperature was 559° F. Find the percentage of available heat carried up the stack by the dry flue-gas.

14. Water is fed to a boiler at a temperature of 170° F. The pressure gage reads 140 pounds, and the barometer 29.6 inches. How many B.t.u. are needed to generate a pound of dry steam? What is the temperature of the steam generated? The volume per pound?

15. Steam at a gage pressure of 135 pounds is generated from water at 120° F. The temperature of the steam is 490° F. The barometer reads 29.3 inches. Find the B.t.u. required to generate one pound of steam.

16. If the temperature of steam in a condenser is 115° F., what is the greatest possible vacuum-gage reading, if the barometer reads 29.16 inches?

17. If water boiling under a pressure of 185 pounds gage is allowed to escape to the atmosphere (as in a boiler explosion), what percentage of its weight turns to steam? What is the ratio of its new volume to the old? Assume that the barometer reading is 29.6 inches.

18. Dry steam leaves a boiler at a pressure of 180 pounds gage and reaches the engine with a quality of 98 per cent, and a pressure of 177 pounds gage. What percentage of its heat contents has it lost in its passage through the pipe? What percentage of its volume? Assume that the barometer reading is 29.43 inches.

19. If one pound of steam of 95 per cent quality at atmospheric pressure is mixed with 8 pounds of water at 70° F., what will be the resultant temperature? Assume that the barometer reading is 29.00 inches.

20. Dry steam enters a turbine at a pressure of 180 pounds gage; leaving the turbine it passes into a condenser in which the vacuum is 27.6 inches (30-inch basis). The quality of steam as it leaves the turbine is 87%. Neglecting all losses, find how many foot-pounds of work may be obtained from each pound of steam that passes through the turbine.

21. A frictionless piston weighing 7000 pounds is placed in a vertical cylinder 10 inches in diameter. Two pounds of water at 70° F. are placed

under the piston. If 800 B.t.u. are added to the water, how far will the piston move? The barometer reads 29.6 inches.

22. If, in Problem 21, 2500 B.t.u. are added to the water, what will be the weight of steam formed? What will be its temperature? How far will the piston move?

23. A sample of steam is taken from a steam line in which the pressure is 150 pounds gage and is led to a throttling calorimeter in which the temperature is 230° F. and the gage pressure is 3 pounds. The barometer reads 29.4 inches. What is the quality of steam in the line?

24. A horizontal water-tube boiler (B. and W. type) has 10 vertical rows of four-inch tubes, 9 tubes to the row. The tubes are 18 feet long, and the steam drum is 24 feet long and 42 inches in diameter. Find the heating surface and the rated horsepower (*a*) by using as heating surface the outside surface of the tubes and one-half the surface of the drum; (*b*) by Rule 3, p. 26.

25. A horizontal return tubular boiler 60 inches in diameter and 18 feet long has 44 four-inch tubes. Find the heating surface and rated horsepower (*a*) by Rule 1, p. 26, (*b*) by Rule 3, p. 26.

26. A Scotch marine boiler-shell is 16 feet 3 inches in diameter and 12 feet long. There are three furnaces, each 43 inches in diameter. The boiler contains three sections of tubes, each section consisting of 110 three-inch tubes 10 feet long. Find the approximate heating surface and the horsepower.

27. A vertical fire-tube boiler (exposed-tube type) has a diameter of 30 inches and a height of 6 feet. The furnace is 25 inches in diameter and 27 inches high. There are 55 two-inch tubes 45 inches long. The normal water level is 10 inches from the top of the tubes. Find the heating surface and rated horsepower by Rule 2, p. 26.

28. In a test of a B. and W. boiler with a hand-fired furnace at the Sewage Pumping Station, Cleveland, Ohio, the following data were taken:

Rated horsepower of boiler	150
Grate surface	27 square feet
Duration of test	24 hours
Steam pressure	156.3 pounds gage
Temperature of feed water	58° F.
Quality of steam formed	99 per cent.
Total weight of coal fired (wet)	15078 pounds.
Moisture in coal	7.5 per cent.
Total weight of water fed to boiler	105100 pounds.

Find:

(*a*) Factor of evaporation.

(*b*) Dry coal per square foot of grate surface per hour.

(*c*) Equivalent evaporation per hour (from and at 212° F.).

(*d*) Equivalent evaporation per hour per square foot of water-heating surface.

(*e*) Boiler horsepower developed.

(*f*) Percentage of rated capacity developed.

29. In the test of Problem 28, the dry coal had a calorific value of 12292

B.t.u. per pound, and the cost delivered at the boiler room was $3.50 per ton of 2000 pounds. Find:

(*a*) Equivalent evaporation from and at 212° per pound of dry coal.

(*b*) Combined efficiency of boiler, furnace and grate.

(*c*) Coal cost per 1000 pounds of equivalent evaporation.

30. In a test of a B. and W. boiler the following data were taken:

Rated horsepower of boiler	508
Grate surface	90 square feet
Duration of test	16.25 hours
Steam pressure	199 pounds gage
Temperature of feedwater	48.4° F.
Superheat	136.5° F.
Total weight of coal fired (wet)	39670 pounds
Moisture in coal	4.22 per cent
Total weight of water fed to boiler	336200 pounds

Find:

(*a*) Factor of evaporation.

(*b*) Dry coal per square foot of grate surface per hour.

(*c*) Equivalent evaporation from and at 212° per hour.

(*d*) Equivalent evaporation from and at 212° per hour per square foot of water-heating surface.

(*e*) Boiler horsepower developed.

(*f*) Percentage of rated capacity developed.

31. The coal in the test of Problem 30 gave the following proximate analysis when dry: volatile combustible, 19 66 per cent; fixed carbon, 75.41 per cent; ash, 4.93 per cent. The cost delivered to the boiler room was $3.75 per ton of 2000 pounds. Find:

(*a*) Equivalent evaporation per pound of dry coal.

(*b*) Combined efficiency of boiler, furnace and grate.

(*c*) Coal cost per 1000 pounds of equivalent evaporation.

32. Is the boiler of Problems 28 and 29 working harder than that of Problems 30 and 31, or conversely? Give the reason for your answer.

33. Find the size of a pop safety-valve with a 45° seat for a 60-horsepower return-tubular boiler which is to carry a gage pressure of 75 pounds. Assume that the maximum evaporation is 5 pounds of water per hour per square foot of water-heating surface, and that the lift of the valve is 1/30 of the diameter.

34. How many 2.5-inch pop safety-valves would one 4.5-inch valve replace, assuming that the lift is proportional to the diameter?

35. How many 3.5-inch pop safety-valves are required for the boiler of Problem 30? Assume the rate of maximum evaporation as 6 pounds of water per square foot of water-heating surface per hour, and that the lift is 1/30 of the diameter.

36. What should be the size of the pop safety-valve for the boiler of Problem 28

(*a*) Computed as in Problem 35?

(*b*) Computed from the P. G. Darling formula? See p. 59.

(*c*) Computed from the city of Chicago formula? See p. 59.

(*d*) Computed from the city of Philadelphia formula? See p. 59.

(*e*) Computed from the U. S. Supervising Inspectors' formula? See p. 59.

(*f*) Computed from the A. S. M. E. Boiler Code Committee's requirements? See Report of Boiler Code Committee of A. S. M. E.

37. What should be the size of a steam pipe leading from a 250-horsepower boiler if the pressure carried is 160 pounds gage? Assume a velocity of flow in the pipe of 5000 feet per minute.

38. A 5000-kw. steam turbine requires 16 pounds of dry steam per hour per kw. at 160 pounds gage pressure. The vacuum in the exhaust of the turbine is 27.5 inches of mercury (30-inch barometer). The quality of steam in the exhaust is 85%. If the velocity of flow of steam to and away from the turbine is to be 7500 feet per minute, what should be the size of steam and exhaust pipes?

39. If a steel steam pipe is to carry steam at a pressure of 200 pounds gage and may be as cold as 30° F. when the steam is cut off, how far apart should expansion joints be placed if each joint gives a 3-inch movement?

40. If 9536 pounds of water at a temperature of 60° F. are mixed with 1160 pounds of steam at 3 pounds gage pressure, the steam being of 90 per cent quality, what will be the resultant temperature of the mixture?

41. The exhaust from a 65-horsepower steam engine is led to an *open* feedwater heater. The engine uses 30 pounds of steam per hour per horsepower, and the quality of the exhaust steam is 80%. The heater is at atmospheric pressure; water enters at 50° F. and is heated to 200° F.

(*a*) What horsepower of boilers will the heater supply?

(*b*) What should be the size of steam and water pipes leading to the heater? Assume a steam velocity of 5000 feet per minute and a water velocity of 150 feet per minute.

42. A 4000-kw. steam turbine is equipped with a surface condenser The turbine uses 16 pounds of steam per kw. per hour, which enters the condenser at a quality of 85 per cent. The vacuum to be maintained is 28 inches (30-inch basis). The circulating water enters the condenser at a temperature of 60° F., and leaves at a temperature 10° cooler than that of the incoming steam. (*a*) How much circulating water is needed per hour?

(*b*) If the same amount of water is circulated as in part (*a*), but if it enters at 90° instead of 60°, and leaves at 10° cooler than the incoming steam, what vacuum can be maintained?

43. An 18″×24″ steam engine has a piston rod 2.75 inches in diameter. Find the head-end and the crank-end piston displacements in cubic feet.

44. If it takes 10.6 pounds of water to fill the head-end clearance space and 11.2 pounds to fill the crank-end clearance space of the engine in Problem 43, what is the percentage of clearance for each end of the engine?

45. Find the volume of steam back of the piston of the engine of Problem 43: when the piston is at 12.4 per cent of the head-end stroke; when it is at 14.0 per cent of the crank-end stroke.

46. Find the weight of dry steam back of the piston of a 24″×36″ engine when it is at 30 per cent of the head-end stroke. The head-end clearance is 4 per cent and the steam pressure back of the piston at the above position is 105 pounds gage. If we know that at this time there is actually 1.06 pounds of wet steam back of the piston, what must be its quality?

47. Construct a hypothetical indicator diagram, using the following data.

Length of diagram = 4 inches (this does *not* include clearance).
Initial pressure = 150 pounds per square inch (gage).
Back pressure = 5 pounds per square inch (gage).
Cut-off = 25 per cent, Release = 95 per cent.
Compression = 15 per cent, Admission = 2 per cent.
Clearance = 7 per cent.
Atmospheric pressure = 15 pounds per square inch.

Use as a scale of pressure 60 pounds per inch.

48. Construct a hypothetical indicator diagram for a uniflow engine (see § 108), using the following data.

Length of diagram = 4 inches. Initial pressure = 170 pounds.
Back pressure = −12 pounds (engine is running condensing).
Cut-off = 20 per cent. Release and compression each = 90%.
Admission = 2 per cent. Clearance = 3 per cent.

Also show, by a dotted line on the same diagram, the compression curve when the engine runs non-condensing (back pressure = 0).

State in what ways this excessive compression may be relieved.

49. Compute approximately the percentage of head-end and of crank-end clearance of the engine from which the cards of Fig. B were taken. Use two methods. Cards were taken with an 80-pound spring.

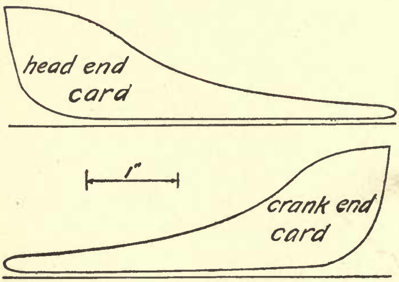


Fig. B

50. Compute the engine constant, or the horsepower constant ($LA/33000$), for the head end and for the crank end for a 10″×14″ engine with a 2″ piston rod. Your answer must be correct to within one-half of one per cent.

51. Find the indicated horsepower (i. hp.) of the steam engine of Problem 50 when the head-end mean effective pressure is 34.2, and the crank-end m.e.p. is 35.4 pounds per square inch. The engine is running at 260 r.p.m.

52. A test was run on a 14″×18″ steam engine with a 2″ rod. The head-end m. e. p. was found to be 35.2 pounds per square inch and the crank-end m.e.p. 34.6 pounds per square inch. The speed of the engine was 250 r.p.m. The power was absorbed by a Prony brake whose arm is 6′ 5″ long. The effective weight of the brake arm on the scales was 45 pounds. During the test the pressure on the scales was 382 pounds. Find (*a*) the indicated horsepower; (*b*) the brake horsepower; (*c*) the mechanical efficiency.

53. The test of Problem 52 was run for 45 minutes, during which time the engine used 2750 pounds of steam at a pressure of 120 pounds gage, and at a quality of 97 per cent. Find:

(*a*) Dry steam used per indicated horsepower per hour.
(*b*) B.t.u. per indicated horsepower per minute.
(*c*) Thermal efficiency based on i. hp.
(*d*) Thermal efficiency based on b. hp.

The indicator diagram in Fig. C and the following data were taken during a test of a Buckeye engine.

Size of engine, $7.75'' \times 15''$, $1\frac{3}{32}''$ rod.

Radius of Prony brake arm = 6.02 feet.

Room temperature = 73.5° F.

Temperature in throttling calorimeter = 221.5° F.

Steam pressure at throttle = 128.7 pounds per square inch gage.

Steam pressure in calorimeter = 1.125 pounds gage.

Barometer = 28.5 inch.

R.p.m. = 222.5.

Net brake load = 140 pounds.

Scale of indicator spring = 80 pounds.

Steam used per hour = 1161 pounds.

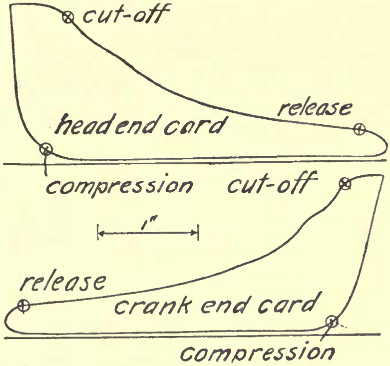


Fig. C

54. Find the m.e.p. of the cards by the mean-ordinate method.

55. Find the indicated horsepower, head-end, crank-end, and total.

56. Find the brake horsepower.

57. Find:

(*a*) Mechanical efficiency.

(*b*) Pounds of steam per i. hp. per hour.

(*c*) B.t.u. per i. hp. per minute.

(*d*) Thermal efficiency based on b. hp.

58. Determine from each card the percentage of stroke and the steam pressure for each of the following events:

(*a*) Cut-off.

(*b*) Release

(*c*) Compression.

59. Determine the weight of dry steam back of the piston for each end at the events of cut-off, release, and compression.

60. Find the amount of re-evaporation or condensation per hour during expansion.

61. Find the weight of dry steam per hour per indicated horsepower accounted for by the cards.

62. Combine the indicator diagrams shown in Fig. D, and determine the diagram factor. The cards of Fig. D were taken from an $8.02'' \times 15'' \times 24''$ cross-compound Corliss steam engine, running at 85 r.p.m. The head-end clearance of the high-pressure cylinder is 7.4 per cent and the head-end clearance of the low-pressure cylinder is 6.01 per cent.

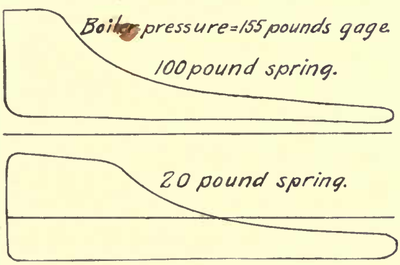


Fig. D

63. Determine the size of cylinders for a compound, two-cylinder, double-acting steam engine (receiver type), assuming the following data: i. hp. = 120, r.p.m. = 100, cylinder ratio = 1/3, piston speed = 600 feet per minute, initial steam pressure = 140 pounds absolute, terminal pressure of hypothetical diagram = 14 pounds absolute, vacuum = 24 inches (30-inch basis), and diagram factor = .85

64. In a certain two-cylinder compound steam engine the number of expansions is 10, the initial steam pressure is 120 pounds absolute and the back pressure is 5 pounds absolute. The receiver pressure is 30 pounds absolute. The cylinder ratio is 1 to 3. Neglecting clearance and piston rods, compare the work done in the two cylinders and the stresses on the two piston rods.

65. Given a cross-compound steam engine, show by means of a graph the variation in power distribution when the governor varies the cut-off equally in each cylinder (choose at least three cut-offs).

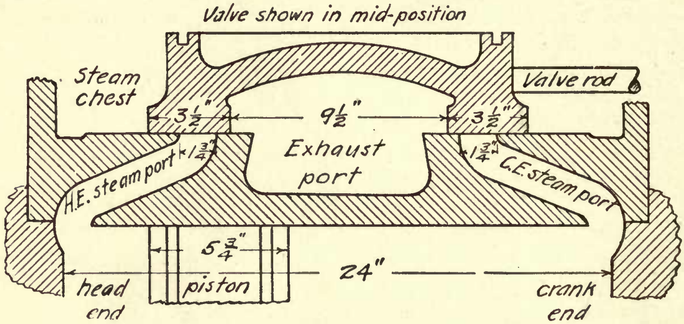


FIG. E

66. Proceed as for Problem 65, but assume that the governor varies the point of cut-off in the high pressure cylinder only.

67. Consider a 12″×18″ steam engine (section of cylinder and valve shown in Fig. E), with the following given data.

Connecting rods 6 feet long.
Valve-travel = 6 inches.
Head-end lead = crank-end lead = .25 inch.
Head-end steam lap = 1.25 inches.
Head-end exhaust lap = .5 inch.
Width of port = 1.75 inches.

(*a*) Draw the valve on its seat, the crank position, the eccentric position, and the position of the piston in the cylinder when the crank is on head-end dead center. (Make your drawing ¼ actual size.)

(*b*) Draw the same parts for head-end cut-off.

(*c*) Draw the same parts for head-end admission.

(*d*) Draw the same parts for head-end release.

(*e*) Draw the same parts for head-end compression.

(*f*) Determine the percentage of the stroke for each of the above events.

68. Consider a 14″×16″ engine, running over, with direct slide-valve and with the following data.

$R/L = 1/6$,
valve-travel = 4 inches,
lead = ¼ inch,
steam lap = 1 inch,
exhaust lap = ¼ inch.

(a) *Valve ellipse.* Draw the crank circle ¼ actual size, and about the same center draw the eccentric circle full size. Choose 12 equidistant crank positions and find the corresponding eccentric positions.

For any crank position (as C, Fig. F_2), the piston is at a distance x from its mid-position, and at the same time the eccentric is at a distance y from its

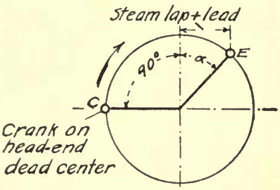


FIG. F_1

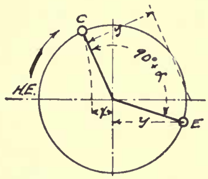


FIG. F_3

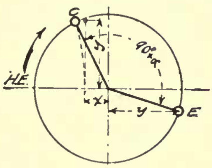


FIG. F_2

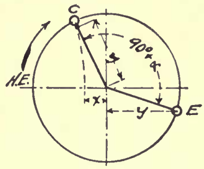


FIG. F_4

mid-position. Plot y vertically and x horizontally, for all 12 positions of the crank.

Connect the points thus found by a smooth curve. Label on this diagram the following details: the crank position at each event of the stroke, the lead, the steam lap, the exhaust lap, the maximum port-openings, and the angle of advance.

(b) *Bilgram diagram.* Draw the crank and eccentric circles and choose 12 equidistant crank positions as in (a). For each crank position (as C in Fig. F_3), draw a dotted line parallel at a distance y from the crank. The intersection of these dotted lines is the Bilgram construction point P. About this point P, draw in the steam-lap and exhaust-lap circles.

Show on this diagram the crank position at each event, the lead, the steam lap, the exhaust lap, the maximum port-openings, and the angle of advance.

(c) *Zeuner diagram.* Draw the crank and eccentric circles as before, and choose 12 equidistant crank positions. Lay off radially on the crank from the center of the crank circle the eccentric displacement y (Fig. F_4); connect all points thus found by a smooth curve.

Show on this diagram the crank position at each event, the lead, the steam lap, the exhaust lap, the maximum port-openings, and the angle of advance.

69. Consider an engine with the following given data.

Direct slide-valve. Head-end steam lap $=1\frac{1}{8}''$.
Engine running over. Crank-end steam lap $=1$ inch.
Valve-travel $=5$ inches. Head-end exhaust lap $=\frac{1}{4}$ inch.
Head-end lead $=1/8$ inch. $R/L=1/5$.

Find the head-end and crank-end crank positions, and the percent of stroke at each event by means of

(*a*) The valve ellipse, (*b*) the Bilgram diagram, (*c*) the Zeuner diagram.

70. Consider an engine with a direct slide-valve and with the following given data:

Engine running over. Crank-end cut-off $=50$ per cent.
Valve-travel $=3$ inches. Head-end compression $=25$ per cent.
Head-end admission $=1$ per cent. Crank-end compression $=25$ per cent.
Head-end cut-off $=50$ per cent. $R/L=1/6$.

Find the percentage of stroke at all events, the angle of advance in degrees, the steam laps, the exhaust laps, the maximum port-openings, and the leads, by means of

(*a*) The Bilgram diagram, (*b*) the Zeuner diagram.

Draw the eccentric circle full size and the crank circle to such a scale that it is the same size as the eccentric circle. Label all of the dimensions asked for directly on the diagrams, also label the head end of the diagram and indicate by an arrow the direction of rotation of the crank.

71. Consider an engine with an indirect slide-valve and with the following given data.

Engine running over.
Valve-travel $=4$ inches.
Head-end lead $=\frac{1}{8}$ inch.
Crank-end lead $=\frac{1}{4}$ inch.
Head-end cut-off $=35$ per cent.
Head-end compression $=15$ per cent.

Sum of steam lap and exhaust lap the same for both ends. $R/L=\frac{1}{4}$.

Find the percentage of stroke at all events, the angle of advance in degrees, the steam laps, the exhaust laps, and the maximum port-openings, by means of

(*a*) The Bilgram diagram, (*b*) the Zeuner diagram.

72. Consider an engine with a direct slide-valve and with the following data.

Engine running over.
Head-end admission $=2$ per cent.
Head-end cut-off $=60$ per cent.
Head-end maximum port-opening $=1.25$ inches.
Crank-end maximum port-opening $=1.25$ inches.
Head-end compression $=20$ per cent.
Crank-end compression $=20$ per cent.
$R/L=1/6$.

Find the valve-travel, the angle of advance, and each of the laps, by means of

(*a*) The Bilgram diagram, (*b*) the Zeuner diagram.

Also draw to scale the valve on the seat in its mid-position.

73. Consider an engine with a direct slide-valve and with the following data.

Engine running *under*.
Head-end lead = $\frac{1}{4}$ inch.
Crank-end lead = $\frac{3}{8}$ inch.
Head-end cut-off = 55 per cent.
Head-end compression = 20 per cent.
Crank-end compression = 20 per cent.
Head-end maximum port-opening = 1.25 inch.
$R/L = 1/5$.

Find the valve-travel, the angle of advance, and each of the laps by (*a*) The Bilgram diagram, (*b*) the Zeuner diagram.
Draw the valve to scale in mid-position.

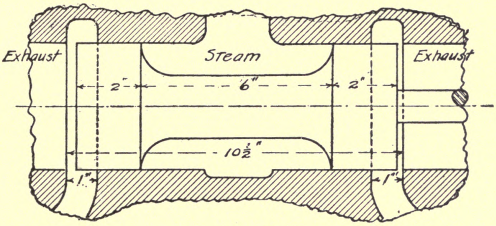


FIG. G

74. Consider an engine with a valve whose dimensions and seat are as shown in Fig. G. The valve is not shown in mid-position. The valve-travel is 4 inches; $R/L = 1/5$; the engine runs over.

The cards of Fig. H are taken with the valve as now set. Find the angle through which the eccentric must be shifted (state whether backward or for-

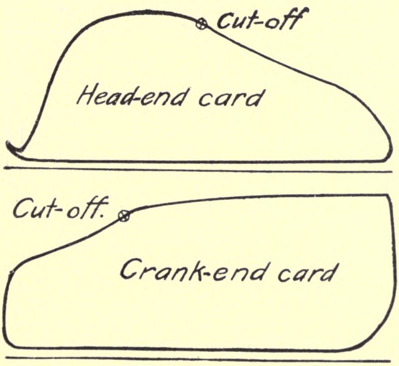


FIG. H

ward), and the amount the valve stem must be lengthened or shortened (state which), in order to give a cut-off of 25 per cent on each end. Draw the approximate cards for the new setting.

75. Consider an automatic shaft governor with the following given data (The Rites inertia governor is shown in Figs. I and J.):

Head-end steam lap = 1.75 inches.
Head-end exhaust lap = 0.
Lead at normal position of eccentric = 5/32 inch.
Distance of eccentric center from pivot (R) = 12″.
Distance from center of shaft to pivot point (x) = $13\frac{7}{8}$″.

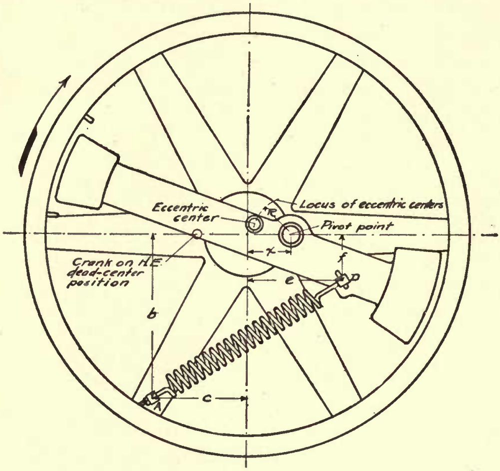


Fig. I

Location of point DE = 25″, f = 18″.
Location of point A: c = 30″, b = 52″.
$R/L = 1/6$.
Cut-off at no load = 10 per cent.
Cut-off at full load = 65 per cent.
Direct slide-valve, engine running over.

I. Draw the governor analysis (full size) and find the valve-travel and angle of advance

(*a*) At 10% cut-off, (*b*) at 65% cut-off, (*c*) at normal cut-off.

(*d*) Also find percent of normal cut-off.

II. Draw the head-end Zeuner diagram, or the Bilgram diagram, for

(*a*) 10% cut-off, (*b*) normal cut-off, (*c*) 65% cut-off.

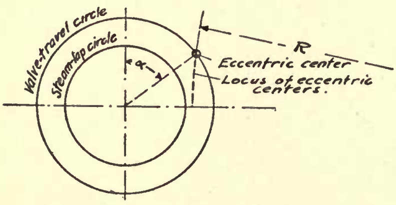


Fig. J

From the events thus determined, draw the theoretical indicator diagrams, using 6 per cent clearance, 150 pounds initial steam pressure, 5 pounds back pressure, and 80 pounds per inch as the scale of spring.

Find the elongation of governor spring (drawing ½ size).

(*a*) From no load to normal, (*b*) from normal to full load.

76. Consider a *four-valve engine,* such as is shown in Figs. 60 and 61, p. 104, with head-end valves as shown in Fig. K_1 and K_2, and with the following data.

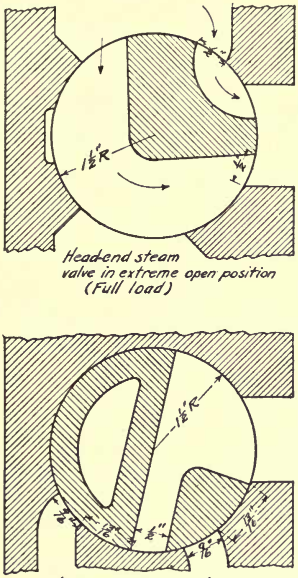
Radius of steam valve arms $= 5''$.

Maximum diameter of steam or cut-off eccentric circle $= 4''$.

Diameter of exhaust eccentric circle $= 4''$.

With the crank on head-end dead center, the pivot point of the governor arm for the cut-off eccentric is on the horizontal center line $8\frac{1}{2}''$ beyond the center of the shaft. Radius of locus of eccentric centers $= 7\ 5/16''$.

Cut-off at maximum load $= 65$ per cent. Compression $= 15$ per cent.
Cut-off at normal load $= 25$ per cent. Release $= 95$ per cent



Head-end steam valve in extreme open position (Full load)

Head-end exhaust valve in extreme closed position

FIG. K

At normal load the steam-valve arm is vertical when the valve is in extreme closed position. The exhaust-valve arm is vertical when that valve is in extreme open position. $R/L = 1/6$. Engine runs over.

Find the angle of advance for each eccentric at normal load.

Find the location of the steam-valve arm when in mid-position, at cut-off, at admission, and when it is in extreme open position at normal load.

Find the maximum port-opening at normal load, and the lead at normal load.

Draw the head-end steam valve in extreme position, and in open position at normal load.

Draw the head-end exhaust valve in extreme open position.

77. The necessary dimensions of a Corliss engine are given in Figs. L and M.

Consider such an engine with the following data.

A 12″×24″ Corliss engine running at 150 r.p.m.
All valves operated by one eccentric.
$R/L=1/6$.
Engine runs over.

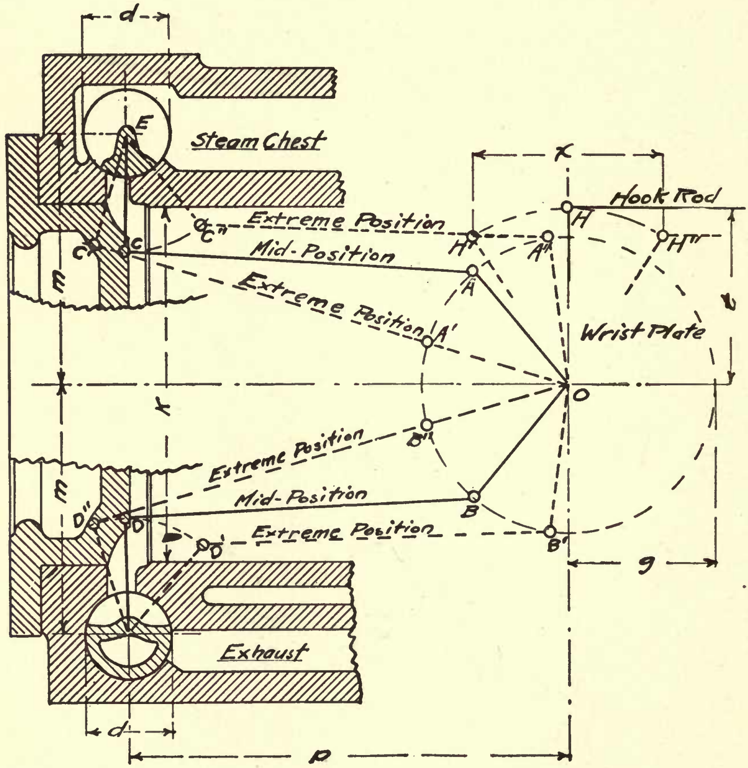


Fig. L

Diameter of all valves = 3″ (d, Fig. L).
m, Fig. L = 8½″;
k, Fig. L = 15″.
Length of valve arms = 4″.
Center of wrist plate is equidistant from all valves.
Radius AO and BO on wrist plate = 5″.
Radius HO on wrist plate (t) = 6″.
Angles $EC'A'$ and $FD''B''=90°$.
Release = 98 per cent.

Compression = 4 per cent.
Crank angle at admission = 3°.
Throw of eccentric = $6\frac{3}{4}$.″
Radii of rocker arms are equal for eccentric and hook rods.
Normal cut-off = 20%.
Width of admission port = $\frac{3}{4}$″.
Width of exhaust port = 1″.
Single-ported valves.

In Fig. M,

Radius of arm EG = $3\frac{1}{2}$′
Radius of arm EH = $4\frac{1}{2}$″.
Radius of arm EI = 4″.
Radius of cam EJ = 2″.
Radius of latch EK = $3\frac{7}{8}$″.
Center G is $1\frac{1}{4}$″ above horizontal center line at trip position for normal cut-off.

Make the general layout $\frac{1}{2}$ actual size, and that of the trip mechanism full size.

Find the lengths of the steam rod AC and the exhaust rod BD, the angle of advance, the steam lap, the lead, the exhaust lap, the maximum

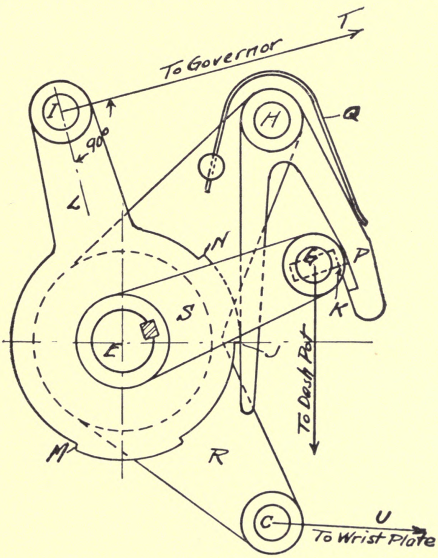


Fig. M

cut-off with trip working, the maximum cut-off when beyond control of trip, the maximum port-opening for maximum cut-off, the maximum port-opening for normal cut-off, the maximum port-opening for 10% cut-off, the movement of the governor rod from normal to 10% cut-off, and the movement of governor rod from normal to maximum trip cut-off.

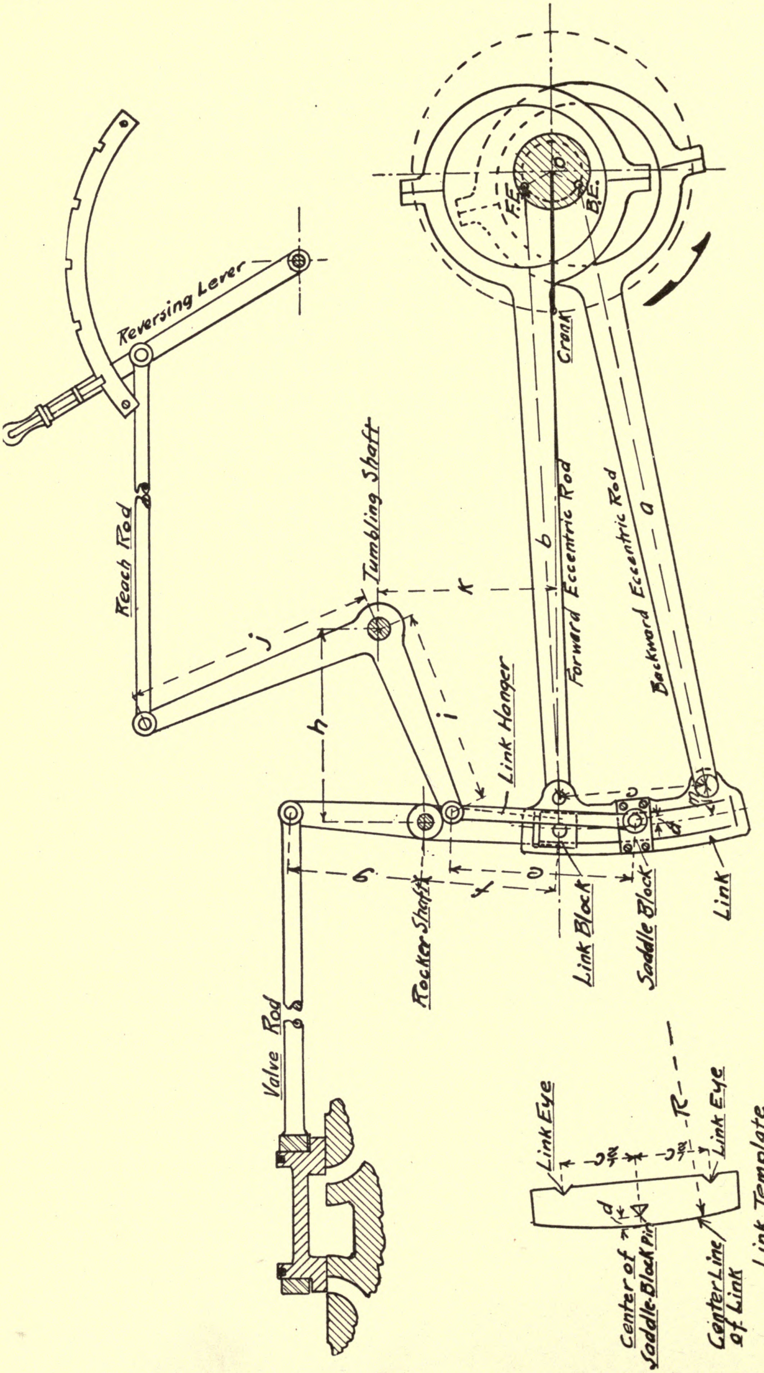


FIG. N STEPHENSON LINK
Shown in Full Forward Position. Crank on Head-end Dead Center

78. The necessary dimensions of a Stephenson link are as follows. (Fig. N.)

Valve-travel at full gear $=5\frac{1}{2}''$.

Steam lap $=\frac{7}{8}''$.

Exhaust lap $=\frac{1}{16}''$.

Lead at full gear $=0''$.

Steam port $=1\frac{1}{4}''$.

Exhaust port $=2\frac{1}{2}''$.

Bridge $=1''$.

$R/L=1/7.5$

$a=b=45''$.

$m=3''$.

$e=14''$.

$g=7\frac{1}{2}''$.

$i=18''$.

$h=14''$.

$c=13''$.

$d=\frac{7}{16}''$.

$f=7\frac{1}{2}''$.

$j=17\ 13/32''$.

$k=18''$.

$R= =48.8''$.

Make the drawing one-half actual size and proceed as follows.

(1) Make a template of the link as shown in Fig. N.

(2) Locate the center of the link-block with the crank at head-end dead center at full gear.

(3) Find the center of travel of the link-block, neglecting for the time being the angularity of the eccentric rods.

(4) Place the center of the rocker shaft above the point found.

(5) Place the crank and eccentrics in their positions at 40% head-end cut-off, running forward.

(6) Find by trial the position of the link (template) for the preceding position of the crank. (Remember the center of the link-block is now at a distance equal to the steam-lap from its mid-position.) Locate the saddle-block pin and the position of the bell crank.

(7) Assume twelve equidistant crank positions and the corresponding eccentric positions for the preceding cut-off. Then draw in the center of link for each crank position, by trial by means of the template.

(8) Draw a valve ellipse from the valve displacement found above. Locate all events.

(9) Check as to the assumed cut-off.

(10) Find the amount of slip between the link-block and link when running at the assumed cut-off.

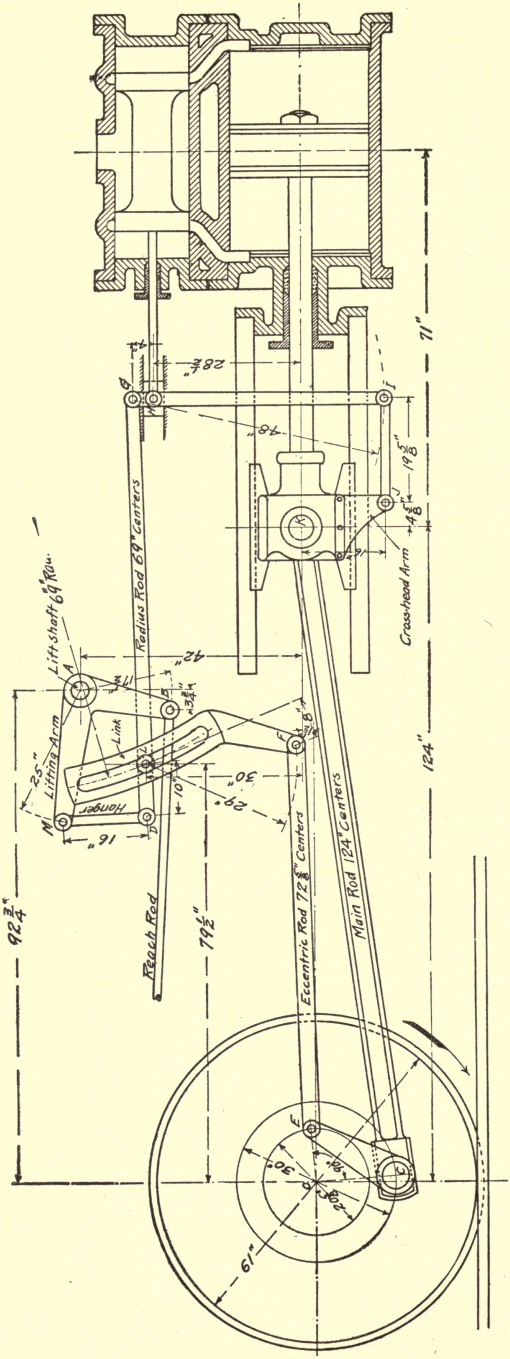


Fig. O Walschaert Valve Gear

79. The arrangement of parts and the necessary dimensions of a Walschaert gear are shown in Fig. O. The valve is of the piston type and has inside admission. This is the common type used on modern locomotives. Fig. O shows the piston in its mid-position and the link-block set at mid-gear. As the link is pivoted to the frame of the engine at the point L, there will be no motion of the link-block when set at mid-gear. Hence all the motion the valve gets at this position of the block comes from the cross-head. Therefore, as the cross-head is in mid-position, the valve will also be in mid-position.

Suppose that such a gear is used on an engine with the following data.

$26\frac{1}{2}'' \times 30''$ engine.
Engine runs forward (under).
Diameter of valve = 14.″
Maximum valve-travel = $6\frac{1}{2}''$.
Steam lap = 1 1/16″.
Exhaust lap = 0.
Lead = 3/16″.
Dimensions as in Fig. O.

Make the drawing one-fourth actual size, and proceed as follows:

(1) Lay out the gear in the position shown in Fig. O, with the piston in mid-position and the link-block set at mid-gear. Indicate each of the parts by its center line only.

(2) Make a template of the link.

(3) Set the valve and the point H for head-end cut-off. Then place the crank at 40% of forward stroke, assuming that the engine is running *forward.*

(4) Locate the eccentric E at 90° back of the crank, and the point F, and draw in the center line of the link.

(5) With the cross-head K in position for 40% cut-off, the location of the point I will be determined. Since H was located in (3), the point G is found by connecting I and H. The distance that G is to the right of the mid-position gives the distance that the link-block center is to the right of its mid-position. This locates the link-block for 40% cut-off. Now locate the points D, M and B.

(6) With the link-block set for 40% head-end cut-off, take twelve equidistant crank positions and find the valve displacement for each position. Plot these valve displacements against the corresponding piston displacements either as in a Zeuner diagram or as in a valve-ellipse diagram, and connect the points thus found by a smooth curve.

(7) Draw in the laps on the diagram just constructed, and check the cut-off with the assumed value of 40%.

(8) Find the amount of slip between the link-block and the link when the position is that of 40% cut-off, with the engine running forward.

80. The Russell Engine Co. makes a four-valve engine. In this engine, the exhaust is taken care of by oscillating or Corliss valves, and the admission by a direct slide-valve. This slide-valve admits steam and carries on its back a rider-valve that cuts off the steam. The main valve is driven by an eccentric keyed to the shaft, while the rider-valve is driven by an eccentric whose angle of advance is controlled by the governor. The governor simply rotates the eccentric about the shaft, changing the angle of

advance, but affecting in no way the absolute travel of the valve. Hence, it is necessary to consider the *relative* motion of the rider-valve and the main valve in making an analysis of the cut-off valve.

The necessary dimensions of the valves and the seat are given in Fig. P. The throw of both eccentrics is 5¼ inches, and the angle of advance of the main eccentric is 32.5 degrees.

Proceed with the analysis in the following manner.

(1) Draw the two eccentric circles about the same center and locate the extremity of the diameter of the valve circle for the main valve.

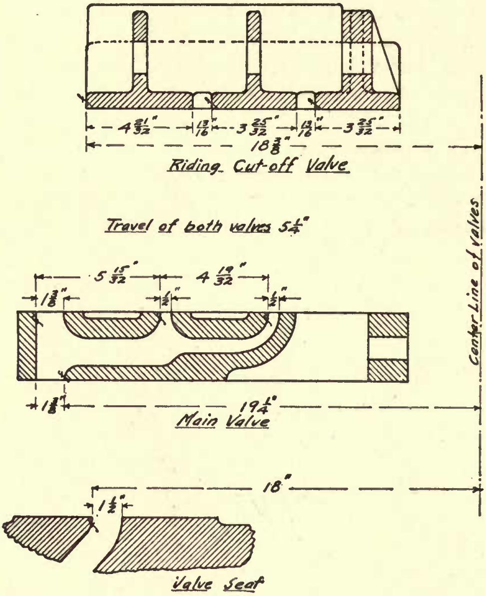


FIG. P · VALVES AND SEAT OF RUSSELL FOUR-VALVE ENGINE

(2) Place the crank for cut-off at 25% of the head-end stroke, and locate the main eccentric. Then determine from the dimensions in Fig. P how far from the mid-position the cut-off eccentric must be to give the proper position of rider-valve for cut-off. This determines the angle of advance of the cut-off eccentric at this particular cut-off.

(3) Take twelve equidistant crank positions and determine the relative position of the rider-valve to the main valve for each position. Plot these distances as in a Zeuner diagram. It will be noticed that the diameter of the relative valve-circle is equal in amount and parallel in direction to a line connecting the extremities of the diameters of the valve circles in the Zeuner diagrams for the main valve and the rider-valve. Also determine the relative steam lap, which is negative. It will be found that this is equal to the distance between the working edges of the rider-valve and the main valve when both are in mid-position.

(4) Now determine the diameters of the relative valve-circles in amount and in direction by repeating the process of (3) for eight cut-offs (0% to 70%), and draw the locus of the extremities of the diameters of the relative valve-circles. It is seen that this locus is the arc of a circle whose radius is equal to the eccentricity of the rider-valve eccentric and whose center is the extremity of the diameter of the valve circle for the main valve.

(5) Make the Zeuner analysis for cut-offs of 10%, 30% and 60%, finding the relative angle of advance and the relative valve-travel by drawing a perpendicular to the crank position at the point where the crank cuts the relative steam lap. The point where this perpendicular intersects the locus of the extremities of the diameters of the relative valve-circles determines the construction point for the relative Zeuner diagram.

81. A gravity-balanced spindle governor built as shown in Fig. Q has arms 20 inches long. At normal speed the arms are at an angle of 45° with the horizontal.

(*a*) Find the normal speed of the governor.

(*b*) Find the percentage of variation in speed from no load to full load.

(*c*) If the normal speed is increased 30 per cent and the range of vertical movement of the point *A* is the same as before, what is the percentage of variation in speed from no load to full load?

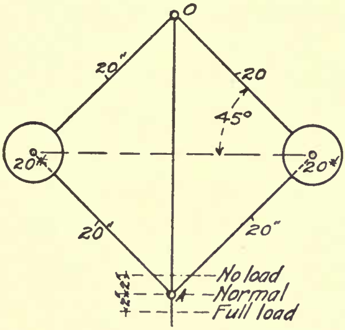


Fig. Q

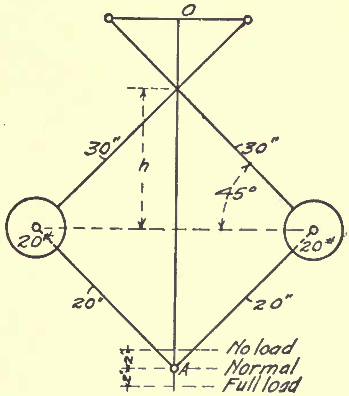


Fig. R

82. In the cross-armed gravity-balanced spindle governor shown in Fig. R, the upper arms are 30 inches long and the lower arms are 20 inches long.

Find the percentage of variation in speed from no load to full load, and compare the result with that of Problem 81.

83. The governor of Problem 81 is now loaded with a weight of 60 pounds. If the normal speed is now 100 r.p.m., and the vertical movement of the point A is the same as before, what is the percentage of variation of speed from normal?

84. A Corliss engine is governed by a loaded gravity-balanced spindle governor. The pulley on the governor and pulley on the engine are both 10 inches in diameter. It is desired to change the speed of the engine from 100 to 120 r.p.m. In what three ways may this be done without affecting the speed regulation? Give your calculations.

85. Find the percentage of variation in speed from no load to full load for the governor shown in Fig. S.

86. Suppose that the spring of a spring-balanced centrifugal governor is fastened directly to a weight of 40 pounds, as shown in Fig. T. If the speed

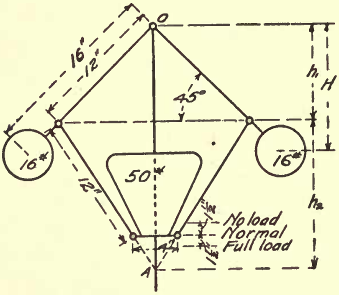


FIG. S

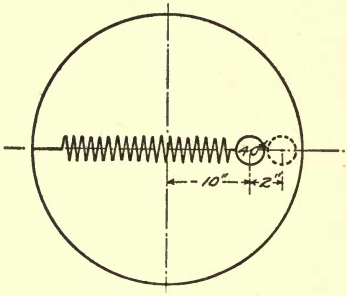


FIG. T

at no load is 206 r.p.m., and at full load 200 r.p.m., what must be the scale of spring?

87. If the spring in Problem 86 is replaced by one of 40-pound scale, and the speed at full load is 200 r.p.m., what is the speed at no load? Is the governor stable or unstable?

88. What scale of spring would make the governor of Problem 86 isochronous at a speed of 200 r.p.m.?

89. The no-load speed of a governor is 300 r.p.m. and the full load speed is 290 r.p.m. At no load, the tension in the governor spring is 500 pounds. At full load, the spring is 2 inches shorter than at no load. The scale of spring is 100 pounds.

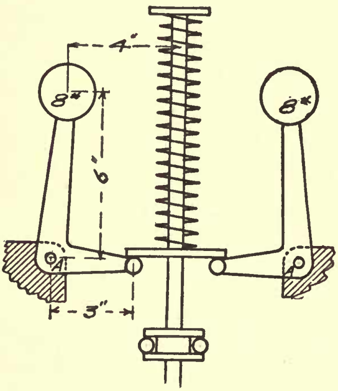


FIG. U

If the spring is tightened by shortening it up half an inch, what will be the effect on the no-load and full-load speeds?

90. With the governor of Problem 89, how much would the spring have to be taken up to make the governor isochronous? What would the speed then be?

91. In the steam-turbine governor shown in Fig. U, the weights of 8 pounds each are 4 inches from the center of the spindle at full load when the speed is 600 r.p.m. At no load, the weights are 5 inches from the spindle and the speed is 610 r.p.m. The weights are pivoted at points A and A.

(a) Compute the scale of spring.

(b) Design the spring, *i.e.* find the size of spring wire, the diameter of the coil, and the number of turns that will give the correct force and scale.

10-12 [illegible]

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